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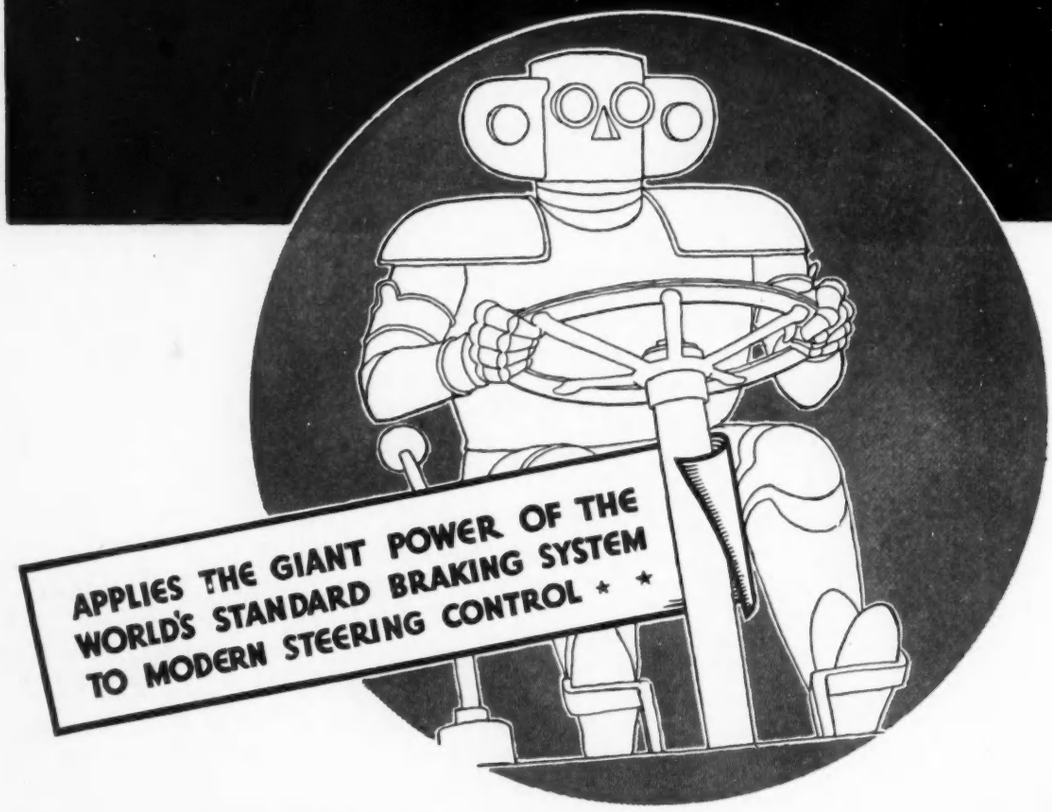
AIR STEERING
another
BENDIX·WESTINGHOUSE
engineering triumph



As a crowning acheivement to years of intense research and development, Bendix-Westinghouse Air Steering Control makes its bow. Easily the most important development in modern motor transport control since the advent of the Air Brake, this unique device offers a positive solution to the rapidly growing steering problem. Built as you would expect Bendix-Westinghouse to build it, Air Steering consists of a compact, light, yet sturdily constructed, double-acting cylinder controlled by a simple valve arrangement and is readily adaptable to any vehicle using a conventional manually-operated steering gear. Ingeniously combining the recognized power and flexibility of air, with just enough physical effort to maintain a desired "wheel feel," Air Steering draws upon the vehicle's existing pneumatic source for the comparatively negligible amount of air required for its operation. Reflecting the traditional features of its lineage, Air Steering assures positive control at all times, even in the remote event of power failure and requires little attention after initial, simple adjustment.

BENDIX-WESTINGHOUSE AUTOMOTIVE AIR BRAKE COMPANY, PITTSBURGH, PA.

Non-Uniform Regulation by States Creates Truck-Building Riddle

By M. C. Horine

BY their lack of uniformity and disregard of scientific and economic fact, legislative restrictions on motor-transport vehicles now in force in the states militate against efficient transportation and thus retard economic recovery. In this indirect way and in several direct ways the same situation presents problems to truck builders. Variations in state requirements necessitate undue diversity of designs, present difficult engineering problems, discourage enterprise, threaten the American system of production and penalize good engineering and sound manufacture.

Although the pertinent facts which should form the basis for equitable legislation of this kind have been determined by the most painstaking scientific research and experiment, under the most responsible auspices and with the endorsement of the highest authorities, and also have been given wide circulation, the states generally seem to have imposed restrictions on motor trucks less for the benefit of the public than to favor selfish interests. Dire results are apt to ensue unless the present oppressive trend of motor-truck legislation is arrested and legislation based upon fact and fairness, uniform as between states, is substituted for the crazy-quilt of present state truck legislation.

TO discuss any subject intelligently and profitably it is first necessary for those discussing it to agree upon a common object. I believe that every sincere student of the broad problem of transportation will agree that the object which all of us seek is economical, convenient and safe transport of persons and property.

No instrumentality of transportation exists for its own sake or for the sake of those who have invested their savings in it for profit. Railroads, motor trucks, canal boats, automobiles, airplanes, motorbuses and animal-drawn vehicles are devised, produced, acquired, operated, patronized and regulated, solely because they contribute to the cheapness, handiness and security of transportation. Viewed broadly, their proper functions are complementary, each to the other, and, under the free play of economic influences, each will inevitably find its own level. Neither regulation, discriminatory taxation nor prejudiced propaganda can change the intrinsic potentialities and limitations of each type of transportation; though these influences can, have, do and will artificially interfere with the economic coordination of transportation facilities, will burden all forms with unnecessary handicaps, reduce the efficiency with which each is used and increase the ultimate cost of transportation to the public.

Certainly, if all interested groups were to hearken to the desires of the public generally and strive for the greatest common good, economy, safety and convenience would be the dominant demand; and if all traffic were permitted to flow unhampered into the channels best adapted in each varied instance to meet this demand, there would be little occasion for vertical competition as between the different agencies of transport. Rather than a continuance of the present senseless efforts of some groups to artificially expand their own sphere by imposing unjust burdens upon the complementary services, all efforts should be bent toward enhancing the efficiency with which each performs its rightful task. Economy certainly, convenience undoubtedly, and safety as well, are promoted just in proportion as efficiency is increased. Artificial restrictions, impositions and discriminations will surely add cost, beget inconvenience and prejudice safety, since they are the implacable enemies of efficiency.

Good transportation is efficient transportation. Inefficient transportation must be more costly, awkward and dangerous. Efficient transportation requires a correctly balanced combination of good operation, good way, good traffic facilities and good vehicles. The importance of good operation is familiar to all. Numerous studies under the auspices of government bureaus and prominent universities have demonstrated the economic advantage of good roads, direct routes, easy curves, moderate grades and the like. Appreciation of the importance of keeping traffic fluid is constantly widening, both for its effect upon transport costs and convenience as well, as it is of paramount importance from the standpoint of safety.

This brings us to the matter of good vehicles. Without good vehicles, the most enlightened operation, perfect routes

[This paper was presented by the Society at the 1933 Highway and Building Congress held at Detroit. Mr. Horine is a Member of the Society and sales promotion manager for the International Motor Co., New York City.]

and smoothest traffic flow would not produce good transportation. But that the engineering brains of the automotive industry have been able to cope successfully with the staggering mechanical, thermodynamic, electrical and other technological problems involved is too overwhelmingly evident to require elaboration here. Unhampered by unwise legislative influence there is not the slightest reason to doubt that whatever the conditions of operation, way or traffic, vehicles adapted to ever more efficient functioning will continue to be produced. But against the uneconomic, unscientific and inequitable meddling of legislative authority, the mighty resources of the automotive industry are relatively inadequate. Efficiency is attained by closest conformance with natural and economic law. Against the vagaries of political law, science is largely impotent.

These immutable laws of nature and mathematics are universal; but man-made law is capricious. Were restrictions on motor trucks a matter of Federal enactment, motor-truck manufacturers could then set about the design of vehicles suitable for the requirements of transport and acceptable to the authorities with confidence and a clear basis upon which to work. If they were predicated, in some measure at least, upon the facts, rather than upon prejudice and misinformation, they could proceed with reasonable assurance of some measure of stability in the demands made of them. As it is, however, there can be no comprehensive planning, for, at every crossing of the 105 imaginary lines separating different sovereign subdivisions of our country, these legislative limitations differ.

With uniform design precluded, the American system of standardized, interchangeable mass production becomes impracticable, and so do coordinated research, development, experimentation, proving and service. In other words, the uncoordinated, arbitrary and contradictory requirements and restrictions of the various states, becoming more divergent and more oppressive as time goes on, bid fair to abolish large-scale manufacture of all but the lighter types of trucks and to reduce truck building from a national industry to a feudal state of local enterprise.

Lest any be moved to regard this inevitable consequence of continuance of our present legislative trend with complaisance, let us not forget that a return to the village wagon-shop type of production will mean restricted engineering, abandonment of research as it is generally practised today and a return to handicraft methods of construction. Such vehicles will be less efficient, much slower to develop, incomparably greater in cost to the purchaser, and their servicing will be a problem quite beyond the skill and ingenuity of the present generation of mechanics. For proof of this, look eastward over the Atlantic, where similar narrow sectionalism has prevented the older and longer experienced European countries from keeping step with the United States in automotive development. Europe does not produce vehicles which are largely marketable in this country, despite longer manufacturing experience and very much lower costs of materials and labor; yet American manufacturers are not only successful in invading the European market but are rapidly displacing long-established European producers in South America, South Africa and Asia.

There are at least a dozen American makes of medium to heavy trucks which can be found in every state, yet in Great Britain, which bears the same economic relation to Europe that a single state of the United States does to our country, nearly all, excepting American makes, are of British manufacture; in Germany they are German; in France, French and so on. The situation in Europe arises from the lack of uniformity in regulation, size and weight restriction, lack of reciprocity between the various states and radically different bases of taxation of both vehicles and their fuel. So surely as the different states in this country further diversify their regulations and restrict reciprocity, the American truck industry will become provincialized as it is in Europe.

49 Problems Face Builders

Legislative restrictions are not *a* problem in truck manufacturing. They constitute 49 problems, since each state and the District of Columbia present individual difficulties to be encountered by the truck builder doing a national business. One often hears the assertion that the vagaries of truck legislation concern chiefly the operator of motor trucks, rather than the producer, since the restrictions in the size of vehicles simply operate to force the more or less fixed volume of traffic into a greater number of smaller trucks, the profit upon whose sale should be the same as on a smaller number of larger types. The case, however, is not so simple as this, for the following reasons:

(1) Since it costs more to move a given volume of freight a given distance on several small trucks than on a few large ones, the tendency of legislatures to force the use of constantly smaller vehicles serves to bring the cost of motor transport up closer to other forms, involving loss of possible motor-transport volume and consequent restriction of the market for motor trucks.

(2) The same influences bring about abnormal competitive conditions which discourage motor-transport enterprises, en-

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"Economy certainly, convenience undoubtedly, and safety as well, are promoted just in proportion as efficiency is increased."

* * * *

"Good transportation is efficient transportation. Inefficient transportation must be more costly, awkward and dangerous."

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courage overloading and under-maintenance and in other ways introduce trade evils which react upon the manufacturer.

(3) Since some states are more liberal than others, the greater preponderance of light chassis does not eliminate the demand for heavier ones at all but merely reduces its volume, thus tending to diversify the sizes and types which manufacturers must produce.

(4) The supposition that there is as much profit in a large number of small trucks as in a smaller number of larger ones is fallacious. The profit which any produce will return to its producer is pretty generally proportional to its earning capacity. It hardly need be stated that the earning capacity of a large truck is greater than that of a smaller one.

Law Changes Uncertain

Transcending all of these aspects of the situation, however, is the effect of the uncertainty and changeableness and the illogical unpredictableness of our truck laws and regulations. The manufacturer can never be sure how long a given product will suit the requirements. For example, over a period of several years, it has been understood that 96 in. was in a fair way of universal adoption as an overall-width limit. In the old days of solid tires it was easy to provide ample tire capacity and still leave room for skid-chain clearance, sprockets and chains, springs and an adequately wide frame. With pneumatic tires, however, particularly the low-pressure type, it has been found extremely difficult to find sufficient room within the 96-in. limit. Owing to their greater deflection, these tires require more center spacing than the high-pressure type and their width is considerable. Every eighth of an inch counts.

With a great deal of difficulty room has been provided for the large sizes required for the heavier trucks by dint of locating the springs under the frame, condensing chain drives and rearranging wheel hubs and brake drums. Realizing these difficulties, some states have permitted up to 108 in. across the tires for change-overs and others have permitted 102-in. widths. In other states, however, not content with the rigid limitation to 96 in., the administering authorities have departed so far from accepted practice and from common sense as to take the measurement over the bulge of the tire where it meets the road. As the larger sizes bulge as much as 1⅜ in. when new and are prone to grow still more with age, this totally unjustifiable practice serves to reduce the practical limitation to 94.5 instead of 96 in. The difficulty and expense which this involves for the manufacturer, who has proceeded in good faith upon the assumption that the width limit of 96 in. as enacted by the majority of the states meant 96 in. in the common interpretation of the term, may well be imagined.

A bulge of 1⅜ in. is less than 1 in. on a side and occurs down at the surface of the road. Normal passing clearance at the widest part of the body is 12 in., so that a bulge of several inches on a side could be of no possible hazard. To keep the overall width of large trucks down to 96 in. has involved such a narrowing of spring centers as to require special means in the way of auxiliary springs to prevent undue side-sway. To further reduce spring centers 1⅜ in. now would prove an intolerable hardship both to users and producers, particularly when necessitated by such unreasonable interpretation of the statutes as is now being practised in some of the states.

Undoubtedly, the greatest confusion is to be found in the statutes themselves. Were the use of motor trucks a new development and were their effects upon highway surfaces, bridges and traffic unknown, there might be a reasonable extenuation for the vagaries of legislation in various states. If climatic differences or differences in topography gave rise to important differences in these effects, then it might be reasonable to expect broad sectional differences. But none of these things are true.

The company with which I am connected has built commercial vehicles exclusively for 33 years. There are other companies nearly as old. Motor-truck operation on a large scale dates back at least to the outbreak of the World War; that is, 16 years. Numerous scientific and economic researches into all phases of motor transport and highway building by the highest and most unimpeachable authorities have been in progress for more than a decade. The findings relate to the physical properties of highways and minima for resistance to climatic effects[1]; impact stresses in pavements produced by different types of vehicles under all conditions; the economic value of highway improvement in relation to traffic volume; analyses of commercial traffic; effects of grades, curves and types of surface on motor-vehicle operating costs[2]; and studies of the fairness of tax distribution as between different types of highway user[3]. All of this mass of definite, accurate and illuminating data is public property and available to all legislative and executive authorities whose duty it is to study it and base their action upon it.

That such procedure has not been followed by any appreciable number of states in framing their laws is proved by the testimony of the laws themselves. The Bureau of Public Roads has stated unequivocally that on hard-surfaced or bituminous-bound roads the destructive potentialities of motor traffic are confined to the impact effect[4] and that this effect is destructive principally in proportion as it is repeated. Traffic studies as well as common sense indicate that the heavier trucks are confined in their operations chiefly to main

[1] See Appendix 1; items (2) and (2a).
[2] See Appendix 2.
[3] See Appendix 1; item (3).
[4] See Appendix 1; item (1).

THE Highway and Road-Building Congress, at which Mr. Horine's paper was presented for the Society at the time of the 1933 Annual Meeting, was sponsored mainly by the American Road Builders' Association and was participated in by some thirty-four allied organizations.

This Congress brought to Detroit the largest number of representatives of the highway and construction industries ever gathered at one time, including Federal, State, county and municipal highway officials, manufacturers of materials and equipment, contractors, distributors, motor-truck operators and manufacturers, insurance company representatives and others whose interests ally them with either highway or general construction enterprise.

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routes, which are generally the best roads. Economic studies have demonstrated that the relatively trifling extra cost involved in building such roads up to a strength capable of withstanding the heaviest practicable traffic results in economies which far outshadow such costs. On the basis of such facts, the Bureau of Public Roads, the Motor Vehicle Conference Committee, the Society of Automotive Engineers and the American Association of State Highway Officials[5] have agreed upon certain fundamentals, as follows:

(1) *Gross Weight,* so far as road surfaces are concerned, is not a measure of road destructiveness. It is of importance, however, as affects the stresses on bridge structures; but in this case its effect varies with the length over which it is applied[6].

(2) *Weight per Inch Width of Tire* is automatically kept within safe bounds by the pneumatic tire, since the tire itself not only tends to increase its contact area as the load is increased but is destroyed by the imposition of unit loads below those destructive to pavements.

(3) *Weight per Axle* is the best measure of road destructiveness. Restriction of axle weights below 16,000 lb. is not warranted scientifically or economically.

(4) *Axle Spacing* is unimportant and should not affect the permissible weight per axle[7] where axles are spaced not closer than 40 in.

(5) *For the Protection of Bridges,* the formula $700 \times (L + 40)$ is adequate[8].

The foregoing are not new recommendations; they have been well known for years and have been incorporated in most of the various codes which have been adopted for some years by various national bodies, designed for legislative guidance. Against them, however, we have the following spectacle: Three states restrict the payload[9]. Thirty-six states and the District of Columbia restrict gross weight[9]; 13 of these permit no greater gross weight on six-wheel than on four-wheel trucks[9] and six permit no greater gross weight on tractor-semi-trailers than on four-wheel trucks[9]. Forty-one states and the District still carry the now obsolete weight-per-inch-of-tire-width restriction. Thirty-seven states and the District have adopted axle-weight limitations[9]. Four of these are below the prescribed minimum-maximum of 16,000 lb., and ten prescribe lower permissible axle loads for six-wheel than for four-wheel trucks.

The facts as to the effects of axle spacing have been ignored by six states in prescribing greater minimum axle spacing than 40 in. for six-wheel trucks and by 14 others which permit greater gross weights on tractor-semi-trailers than on six-wheel trucks. The bridge formula is used in one form or another by seven states, one of which also prescribes gross-weight limitations; of these seven, only four preserve its original form, and all but one have chosen to lower the limits[9]. Of all the 48 states and the District of Columbia, not one has conformed to the findings of scientific authority[10]. From the indications so far apparent, taking Governor Pinchot's famous letter to the governors of all of the other states as an example, there seems to be little disposition to do so.

Aside from the effects of legislative restrictions, manufacturers find themselves mightily affected by taxation policies. Excessive gasoline taxes have exaggerated the importance of fuel cost. A vehicle can be built for economy of fuel as the chief object; but the maximum economy must be achieved by the sacrifice of both performance and mechanical sturdiness. A small, high-speed engine with a multi-speed transmission and a chassis so lightened that its durability is limited, its reliability precarious and its maintenance costs excessive, will show gratifying savings in fuel cost per mile. Going to extremes along this line will naturally result in a long-run total cost per ton-mile which is excessive; but with fuel costs artificially raised by burdensome taxes, some sacrifice in this direction is bound to be encouraged. It is thus seen that high gasoline taxes penalize sound engineering in truck design because they penalize weight, power and durability. There can be little doubt that the present interest in Diesel engines has been greatly intensified by the great contrast in fuel cost offered, largely due to the fact that Diesel fuel is tax-free and because retail distribution costs have not as yet been incurred.

In many of our states a left-handed method of banishing heavy trucks from the road has been employed in license fees which progress geometrically with size. Some states impose taxes on a basis of manufacturer's rated capacity; some on chassis weight; some on gross weight; and others on an assumed gross weight which is an arbitrary figure based on chassis weight. Each of these bases is fallacious and unfair; but when all are taken together they are intolerable. A given chassis in one state is taxed on its nominal rated capacity, regardless of its weight or the actual loads carried. The same chassis in another state is taxed according to its gross weight, regardless of how much of that weight is useful load and how much is tare. Both laws encourage overloading, for, by overloading a truck of small rating, the operator may carry his loads with a smaller tax burden under either scheme.

In 34 states, in addition to axle-weight restrictions, there are parallel restrictions on weight per wheel, these being exactly half the weight permitted on the axle. This at first seems an innocent, though somewhat stupid duplication. On closer analysis, however, it is revealed as a particularly mischievous joker, for, with a certain total weight upon an axle,

[5] Complete text available, but omitted herein.
[6] See Appendix 1; item (1).
[7] See Appendix 1; item (4).
[8] Complete text available, but omitted herein.
[9] Detailed data sheets omitted herein.
[10] Complete text available, but omitted herein.

the weight on individual wheels will vary considerably according to the distribution of load within the body and also upon the transverse inclination of the road. Should the state police decide to do so, they could actually effect a reduction of axle loads permitted by taking pains when weighing a truck upon the road to place the Loadometer under the right rear wheel, selecting for the test a spot where the road crown is greatest or where the shoulder upon which the right rear wheel rests is lower than the pavement. Under these conditions an 8000-lb. wheel-load limitation may easily become a 15,000-lb. axle limit.

Small wonder it is, therefore, that the truck manufacturer finds himself confronted with a different problem in each state within which he does business. The larger the unit which he builds, the fewer places he can find in which to sell it. There has been much idle talk about the wane of the big truck, as though, in these modern times, loads were getting lighter. The fact is that the economic demand for heavy trucks is greater than ever; but the opportunities for using them have been artificially restricted by legislation back of which there is neither reason nor justice.

Monstrosities Born

Undoubtedly, much of technical benefit has resulted from the struggle of vehicle manufacturers to supply their customers with economical units. The tractor-semi-trailer, the full trailer, aluminum bodies and tanks, the six-wheeler and even redistribution of weights on four-wheelers, have all been intensively developed as a result of legislative encumbrances upon motor transport. But, because much of this development has been under duress and not an orderly, economic-engineering development, monstrosities have been given birth.

Overly restrictive axle-load limits have resulted in the construction of four-wheel-truck chassis with long wheelbases and short bodies in order to throw more of the gross weight upon the normally lightly loaded front axles. Developments are now proceeding trending toward a return to the old camel-back models of 20 years ago, with the engine under the cab, in order to carry more of the weight over the front axle and to keep within length limitations with long bodies or semi-trailers. Makeshift tandem-axle attachments have been hurriedly developed to hold the fort while properly engineered six-wheelers were being developed. Everywhere, undersized tractors have been adapted to heavy semi-trailers, without adequate brakes, electrical equipment or strength of driving parts. These under-powered units obstruct traffic by proceeding at a snail's pace wherever grades are encountered; they endanger other users of the highway through inadequate stopping ability and their frequent mechanical failures and short life bring motor transport into ill repute.

Meanwhile, efforts to build and use properly designed, carefully built, adequately equipped units are everywhere being discouraged, ostensibly to protect the roads, but actually to bequeath some fancied benefit upon the railroads, to clear the roads for joy-riders or in a fruitless effort to restore the plundered exchequers of our states. For example, let us take the case of a 5-ton truck of the following specifications: Payload, 5 tons; gross weight, 20,000 lb.; weight on one axle, 14,400 lb.; tires, 9.00-20 balloon; overall width, 89⅞ in.; overall length, 21 ft.; wheelbase, (*L*), 15 ft. Such a vehicle would be outlawed in six states as shown in Table 1: Florida, Kentucky and Vermont, on account of its gross weight; Florida, also because of its overall width; Louisiana and Texas because of its payload; and Mississippi because of its axle load.

[11] Detailed data sheet omitted herein.

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* * * *

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If we build a six-wheel truck for this load, increasing the gross weight to 22,000 lb., reducing the axle load to 8800 lb. and decreasing the overall width to 83⅜ in., it could operate in Mississippi but would be outlawed in all of the other states mentioned and, in addition, Alabama, New Hampshire, North Carolina, Oklahoma, South Dakota and Tennessee would bar it on account of gross weight.

Attempting to carry the same payload on a tractor-semi-trailer with a gross weight of 21,000 lb., an overall width of 79⅞ in., an overall length of 28½ ft. and a total wheelbase, (*L*), of 24 ft., we would still be outlawed in seven states; namely, Alabama, Florida, Kentucky, Oklahoma and Vermont on account of gross weight, and in Louisiana and Texas on account of payload. Similarly, as shown in Table 1, we would find outlawed in

40 states, the 7½-ton four-wheel truck
19 states, the 7½-ton six-wheel truck
10 states, the 7½-ton tractor-semi-trailer
All states, the 10-ton four-wheel truck
32 states, the 10-ton six-wheel truck
35 states, the 10-ton tractor-semi-trailer
All states, the 15-ton six-wheel truck
47 states, the 15-ton tractor-semi-trailer

By the use of full trailers it is possible to carry 5 tons in Florida[11] and Mississippi, which ban the six-wheel truck and one of which bans both the six-wheeler and the semi-trailer. Ten-ton loads may be carried by the use of full trailers in New Hampshire, North Carolina, Oklahoma, South Dakota and Tennessee, although these states would prohibit 5-ton six wheelers and Oklahoma 5-ton tractor-semi-trailers as well. Alabama and Kentucky, however, forbid the operation of full trailers.

In normal times the manufacture of motor trucks is a half-billion dollar industry with an invested capital of about 400 million dollars and is an integral part of our largest industry. The United States is each year investing 1 billion,

Table 1—TRUCK SIZES AND TYPES AFFECTED BY STATE RESTRICTIONS

(Symbols: T-S-T = Tractor and Semi-Trailer; A = Axle Load; F = Formula; G = Gross Weight; L = Length; P = Payload; W = Weight.)

STATE	5-Ton			7½-Ton			10-Ton			15-Ton	
	Four-Wheel	Six-Wheel	T-S-T	Four-Wheel	Six-Wheel	T-S-T	Four-Wheel	Six-Wheel	T-S-T	Six-Wheel	T-S-T
Alabama	.....	G	G	G	G	G	G	G	G	GW	G
Arizona	.....	.....	.....	GA	A	.....	GA	G	G	GAW	ALF
Arkansas	.....	.....	.....	A	.....	.....	A	.....	A	AWF	GA
California	.....	.....	.....	GA	.....	.....	GA	G	G	GAW	G
Colorado	.....	.....	.....	.....	.....	.....	GA	.....	.....	GAW	A
Connecticut	.....	.....	.....	.....	.....	.....	G	.....	.....	G	G
Delaware	.....	.....	.....	A	.....	.....	GA	.....	A	GAW	GA
Florida	GW	G	G	GW	GW	GW	GW	GW	GW	GW	GW
Georgia	.....	.....	.....	A	.....	.....	A	.....	G	GAW	GA
Idaho	.....	.....	.....	GA	.....	.....	GA	.....	A	GAW	GA
Illinois	.....	.....	.....	GA	.....	.....	GA	.....	A	GAW	GA
Indiana	.....	.....	.....	AW	.....	.....	A	F	AF	AWF	AF
Iowa	.....	.....	.....	A	.....	.....	AF	F	AF	AWF	AF
Kansas	.....	.....	.....	A	.....	.....	GA	G	.....	GAW	GA
Kentucky	G	G	G	G	G	G	G	G	GL	GWL	GL
Louisiana	P	P	P	P	P	P	P	P	PG	PW	PG
Maine	.....	.....	.....	GA	G	G	GA	GA	G	GAW	GA
Maryland	.....	.....	.....	.....	.....	.....	GW	W	W	GW	GW
Massachusetts	.....	.....	.....	.....	.....	.....	G	.....	.....	GW	G
Michigan	.....	.....	.....	A	A	.....	A	A	.....	AW	A
Minnesota	.....	.....	.....	A	A	.....	A	A	.....	AW	A
Mississippi	A	.....	.....	GA	GA	G	GA	GA	GA	GAW	GAL
Missouri	.....	.....	.....	GA	G	.....	GA	G	GA	GAW	GA
Montana	.....	.....	.....	GA	.....	.....	GA	GA	G	GAW	GA
Nebraska	.....	.....	.....	A	.....	.....	PA	PG	PGA	PAW	PGA
Nevada	.....	.....	.....	.....	.....	.....	G	.....	.....	GW	G
New Hampshire	.....	G	.....	GA	GA	.....	GA	G	GA	GAW	GA
New Jersey	.....	.....	.....	.....	.....	.....	G	G	.....	GW	.....
New Mexico	.....	.....	.....	A	.....	.....	A	F	F	AWF	AF
New York	.....	.....	.....	.....	.....	.....	A	.....	.....	GAW	.....
North Carolina	.....	G	.....	GW	G	.....	GW	GW	W	GW	GW
North Dakota	.....	.....	.....	A	.....	.....	A	.....	A	AW	A
Ohio	.....	.....	.....	GA	G	.....	GA	G	.....	GAW	A
Oklahoma	.....	G	G	GAW	G	G	GAW	GW	GAW	GAW	GAW
Oregon	.....	.....	.....	A	.....	.....	A	.....	.....	AW	GA
Pennsylvania	.....	.....	.....	A	.....	.....	GA	.....	G	GAW	GA
Rhode Island	.....	.....	.....	.....	.....	.....	GA	.....	.....	GA	G
South Carolina	.....	.....	.....	GAW	GA	G	GAW	GAW	GAW	GAW	GAW
South Dakota	.....	G	.....	GA	G	.....	GA	G	G	GAW	GA
Tennessee	.....	G	.....	GA	GA	.....	GA	GA	G	GAW	GA
Texas	P	P	P	P	P	P	P	P	P	PW	P
Utah	.....	.....	.....	A	.....	.....	GA	G	G	GAW	GA
Vermont	G	G	G	G	G	G	G	G	G	GW	G
Virginia	.....	.....	.....	GA	.....	.....	GA	G	GA	GAW	GA
Washington	.....	.....	.....	G	.....	.....	GA	GA	.....	GAW	GA
West Virginia	.....	.....	.....	A	.....	.....	A	.....	A	AWF	AF
Wisconsin	.....	.....	.....	G	.....	.....	GA	.....	.....	GAW	GA
Wyoming	.....	.....	.....	A	.....	.....	A	F	F	AWF	AF
District of Columbia	.....	.....	.....	.....	.....	.....	G	.....	G	GAW	GL
All Right	43	38	42	9	30	39	0	17	14	0	2
No Good	6	11	7	40	19	10	49	32	35	49	47

700 million in highways and 500 million in streets for vehicular traffic, to which might be added other huge sums represented by complementary equipment owned by operators. The only way in which the public may receive a return on this colossal investment is through the safe and economic utilization of the manufacturing, operating and highway investments made. Government has no higher function than the safeguarding of the safety and well being of its citizens.

By these same means and only so can motor-truck manufacturers thrive. The chief arraignment of motor-truck legislation, therefore, is not its effect upon the narrow interests of particular manufacturers, but rather that broader influence which it has upon national prosperity as so greatly affected by the cost and convenience of motor transport. Let us have legislation which is good for every man, which brings the necessities and luxuries most quickly, easily and cheaply to the door of the common man; with that, motor-truck manufacturers will be content.

"Under-powered units obstruct traffic by proceeding at a snail's pace wherever grades are encountered; they endanger other users of the highway through inadequate stopping ability and their frequent mechanical failures and short life bring motor transport into ill repute."

* * * *

"Efforts to build and use properly designed, carefully built, adequately equipped units are everywhere being discouraged, ostensibly to protect the roads, but actually to bequeath some fancied benefit upon the railroads to clear the roads for joy-riders or in a fruitless effort to restore the plundered exchequers of our states."

* * * *

"Let us have legislation which is good for every man, which brings the necessities and luxuries most quickly, easily and cheaply to the door of the common man; with that, motor-truck manufacturers will be content."

Appendix 1

Docket No. 23,400 of the Interstate Commerce Commission on Coordination of Motor Transportation.—Testimony of Thomas H. McDonald, Chief, U.S. Bureau of Public Roads, Washington, D. C., March 5, 1931:

Question.—"Would a truck of a greater gross load than 23,000 lb. do more damage to the highway than one of 23,000 lb. or less?"

Answer.—(1) "If all other conditions were equal, yes sir. I tried to bring out the point that we are not particularly interested in the gross load. The gross loads do affect bridge construction, but we are not particularly interested in highway designs, in gross loads, but in actual or wheel loads. If a truck of a greater gross load was so balanced by the manufacturer that it had not to exceed 18,000 lb. on the rear wheels, it would do no more damage. We do not talk in terms of gross loads as affecting a design of highways.

(2) "We would not build roads much less than 7 in. thick at the edge and 6 in. in the center, no matter what kind of loads we were going to carry. If we built thinner surfaces they would curl up like tissue paper in the rays of the sun. They would warp; the frost heave would destroy them. So we have a certain minimum thickness of road that it is necessary to build if there were nothing heavier than the ordinary passenger cars and farm trucks to use the road, and the whole question of the heavier buses and heavier trucks therefore begins with a certain minimum thickness of road which is necessary regardless of whether they existed or not.

(2a) "We run up to the 2-ton truck with a 4400-lb. wheel-load and an impact of 7900 lb. and find that this same dimension road would have a stress of 280 lb. The same road would carry a 3-ton truck with 320-lb. stress—well within the working limits of the material which we use at 350 lb. It is not until we get to the 5-ton truck that it is necessary to increase the dimensions of the road and then only by ½ in. in thickness; so that, as between the passenger cars and the lighter trucks—farm trucks, if you wish, taken as a value of one or as a unit—when you get to the 5-ton truck, in order to keep our stress, get the working stress of the material, 350 lb., we must increase the thickness of the road ½ in. or about 7.7 per cent and for a 7½-ton truck, 1.154 in. or about 15.4 per cent.

(3) "In my judgment, the heavier trucks and buses, by the higher tax which they are paying and particularly through the collection of gasoline taxes, are fully meeting all excess costs of construction due to the increased thickness that is made necessary by these heavier loads.

(4) "Our tests show that in the application of wheel loads to the road, if the wheels rest as much as 36 to 40 in. apart, if the point of contact of one wheel is 36 in. ahead of the next wheel, there is no overlap of stress in the road structure. In other words, the road has to carry the weight of each particular wheel. The stresses do not pile up. Therefore, if we had a load to move on the roads that would take more than 18,000 lb. on the rear axle, it should be solved by placing two 18,000-lb. rear axles with four wheels in place of two wheels.

(5) "Wherever the utilization of trucks or buses runs high, we have a concentration of population which uses private vehicles more."

Appendix 2

Bulletin No. 91 of the Engineering Experiment Station, Iowa State College, Ames, Iowa, on Operating Cost Statistics of Automobiles and Trucks.—By Prof. T. R. Agg and H. S. Carter, July 25, 1928:

"It appears, from the statistics included herein, that the average cost of operating an automobile throughout the year on all sorts of roads and under the usual weather conditions varies, according to the weight of the vehicle, from about 6 cents to about 9.5 cents a mile and that the cost for individual automobiles may be slightly lower than the above average or very much higher. An estimate is included which indicates that it costs 0.99 cents a mile more to operate on intermediate-type roads than it does on high type, and about 1.07 cents a mile more to operate on low-type roads than on intermediate type."

New Features in Shock Absorbers With Inertia Control

By C. H. Kindl

MANY improvements in shock-absorbing apparatus have been made during the last two years, the most notable being in dash control and devices for temperature compensation. Two types of hydraulic absorbers, the piston and the vane types, have been in use during this period. Both constructions function around the hydraulic principle of forcing a fluid through an orifice of some type. So-called automatic shock-absorbers were much heard of during 1932. The various kinds of control used are examined herewith to determine whether or not some particular type of velocity-load diagram is most desirable.

After describing the inertia-controlled shock-absorber, Mr. Kindl enlarges upon its various features. The equipment used for testing purposes is illustrated. In conclusion, he states that future experimental work undoubtedly will increase the perfection of this type of shock absorber. Present designs result in outstanding improvements in rides, but further improvements must come as a result of lower spring-rates and lighter wheel-suspensions.

ABOUT five years ago hydraulic shock-absorbers were introduced as standard equipment on several makes of automobiles. The following year virtually all American car manufacturers furnished hydraulic shock-absorbers as standard equipment. This was the beginning of an intense interest in the riding qualities of an automobile, and many improvements in shock-absorbing apparatus have been made during the last two years. The most notable advances have been made in dash control and devices for temperature compensation.

Two types of hydraulic absorbers have been in use during this period; that is, the piston and the vane types. Both constructions function around the hydraulic principle of forcing a fluid through an orifice of some type. Both types have

[This paper was presented at the 1933 Annual Meeting of the Society.
Mr. Kindl is chief engineer of the Delco Products Corp., Dayton, Ohio.]

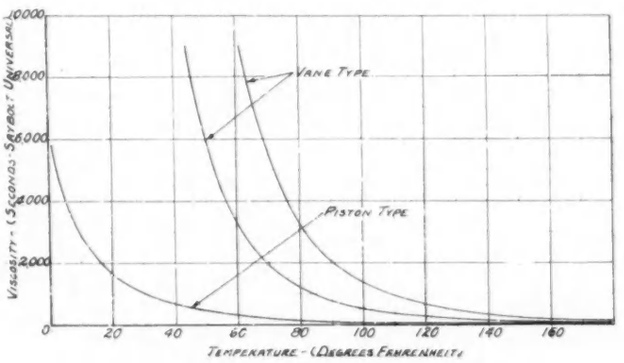


Fig. 1—Temperature-Viscosity Curves of Various Shock-Absorber Fluids

comparative advantages and disadvantages. The vane type using rotary motion is a simple mechanical construction, since the vane is part of the shaft and rotates therewith. The piston type requires a cam or lever connection between the rotating shaft and the reciprocating piston, and this connection must be noiseless and durable. However, the fluid-leakage paths of the piston type can easily be made of considerable length and no difficulty is experienced in grinding

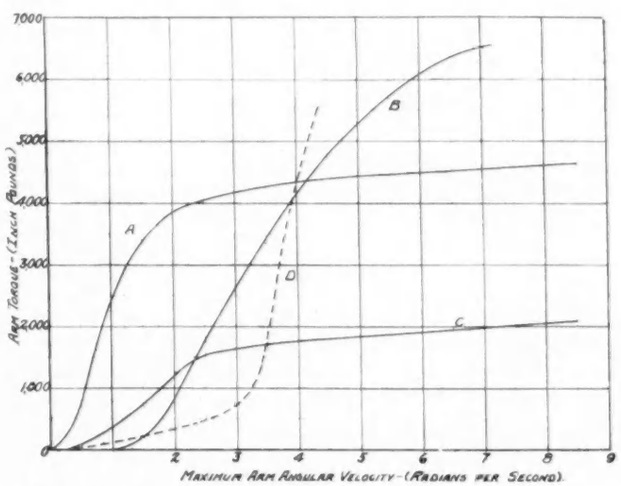


Fig. 2—Performance Curves of Various Shock Absorbers

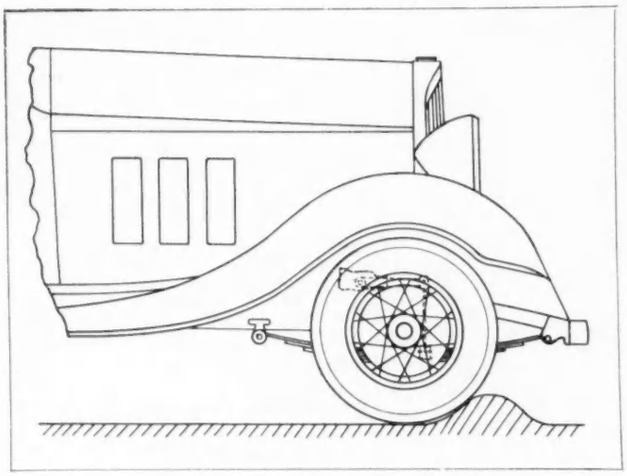


Fig. 3—Diagram Illustrating Why It Is Difficult to Obtain Sufficient Control for High-Speed Operation and at the Same Time Give a Good "Boulevard Ride"

the piston diameter or in boring the cylinder to obtain close fits. The vane-type absorbers inherently have shorter leakage paths and introduce difficult machining problems in maintaining close fits. In view of the foregoing, the vane-type shock-absorber must inherently use a fluid of comparatively high viscosity to compensate for the large leakage, while the piston-type absorber may use a lighter fluid. This is important from the standpoint of temperature effect.

Viscosity curves of the fluids used by two vane-type shock-absorbers and one piston-type are shown in Fig. 1. The viscosity is plotted in rectangular coordinates to more easily show the comparisons. Since the resistance to flow through an orifice is a function of the viscosity, some type of temperature adjustment is necessary for the shock-absorber using a heavy fluid with a fixed-orifice control. Bimetal thermostats are in use at present to change the orifice size for various temperatures, with the object of obtaining more constant resistance regardless of temperature. These devices undoubtedly decrease the effect of temperature but are somewhat delicate in operation, as only a small force is available to vary the orifice size and any slight friction interferes with their operation. The piston-type shock-absorber uses a light oil that is less subject to temperature variation and, in addition, a relief valve controls the shock-absorber resistance independently of temperature or viscosity.

During 1932 much was heard of "automatic" shock-absorbers, and it is of interest to examine the various kinds of control used, to determine whether or not some particular type of velocity-load diagram is most desirable.

Fig. 2 illustrates shock-absorber performance-diagrams and shows the shaft torque with various shaft velocities. Torque is plotted vertically in pound-inches, while the angular velocity of the shaft is plotted in radians per second. The three curves shown in solid lines give performances of shock-absorbers used on three makes of cars of about the same weight. The curves were obtained after adjustments were made to give the kind of ride approved by the car engineers. Obviously, the use of shock-absorber (*c*) on car (*a*) would not give a satisfactory ride. An investigation conducted along these lines shows definitely that no optimum curve for conventional shock-absorber performance exists. The dash-line curve (*d*) shows a performance in which the control increases rapidly after a certain velocity is reached. This type of performance has been found undesirable. The term "automatic" has been rather indefinitely applied to shock-absorbers during the last year. It implies that the shock-absorber adjusts itself

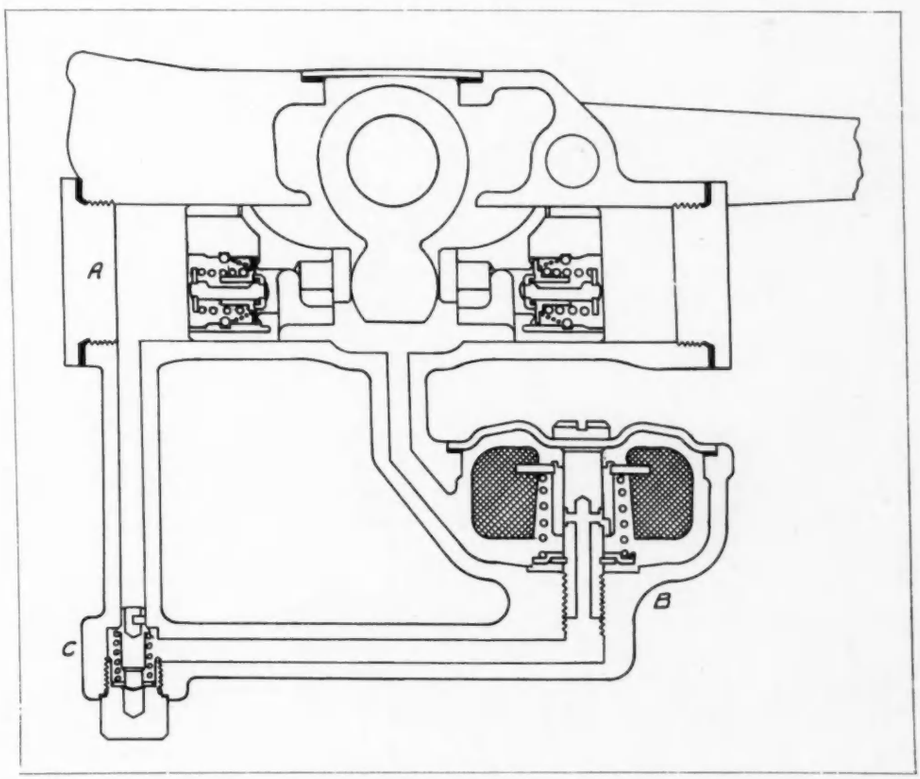


Fig. 4—Diagrammatic Sketch of the Inertia Shock-Absorber

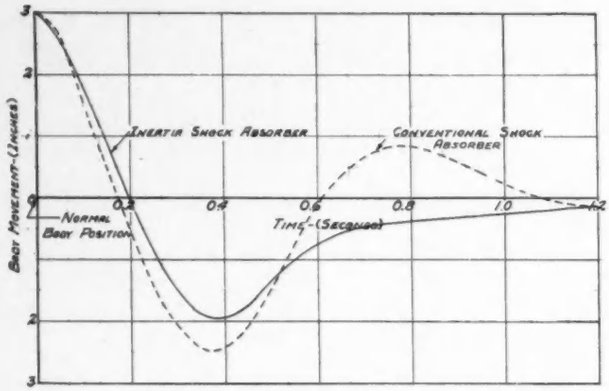


Fig. 5—Body Movement-Time Curve. Rear of Car Dropped 3 In.

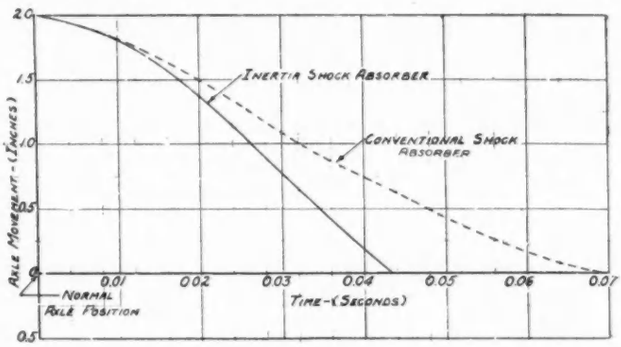


Fig. 6—Axle Movement-Time Curve. Front of Car Dropped 2 In.

for various road and speed conditions; but this is true only in the sense that the control varies as some function of the shaft velocity.

Despite the work done on shock-absorber controls, the establishing of adjustments in working out a ride always has been a compromise. How to obtain sufficient control for high-speed operation and at the same time give a good "boulevard ride," always has presented a problem. The reason for this becomes clear by referring to Fig. 3. As the wheel passes over a ridge in the pavement, the axle moves toward the car body, compressing the springs. Any compression control in a shock-absorber only adds to the spring forces that tend to accelerate the car body upward. After the wheel passes over the obstruction, it moves away from the car body as the springs react. Any fixed rebound control in the shock-absorber tends to prevent the return of the wheel to the road and, as a result, the car body moves downward still farther than if the shock-absorber did not function.

The foregoing phenomena produce undesirable harshness, especially when the shock-absorbers are set for a stiff adjustment. The fact should be remembered that the axle movements discussed usually are of higher velocity than body movements and thereby produce shock-absorber forces that are much greater than are required to control body movements. These forces must be reduced to the minimum to obtain improved rides. This would require an instrument that would distinguish between body and axle movements so that suitable independent controls could be obtained. This consideration led to the development of the inertia-controlled shock-absorber.

A diagrammatic sketch of the inertia shock-absorber is reproduced in Fig. 4. The inertia control is placed in the rebound end (*a*) of the shock-absorber. It consists of a weight, of approximately 1½ lb., balanced by a spring of low rate, at (*b*). In the balanced position, the valve-stem connected to the weight offers no resistance to the passage of fluid. The valve is designed so that the velocity of oil flow through it has no force tending to change the position of the valve; that is, to open or close it. With the weight in the balanced position, the shock-absorber arm can move with minimum resistance. This minimum resistance is determined by the valve (*c*), which is in series with the passage to the inertia-weight valve. Placing some resistance in this circuit is desirable to obtain additional stability of the car body. This is of importance in preventing body roll and slow movements wherein acceleration values are so slight that the inertia control does not function.

The spring of valve (*c*) usually is designed to provide from 20 to 50 lb. of link pull. This value of pull does not produce harshness in the ride and is somewhat equivalent to a definite constant spring-leaf friction. Axle dance is damped by this fixed resistance.

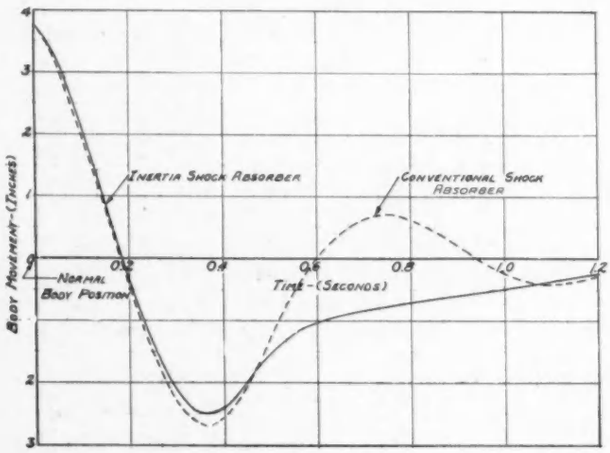


Fig. 7—Body Movement-Time Curve. Rear of Body Dropped 3¾ In.

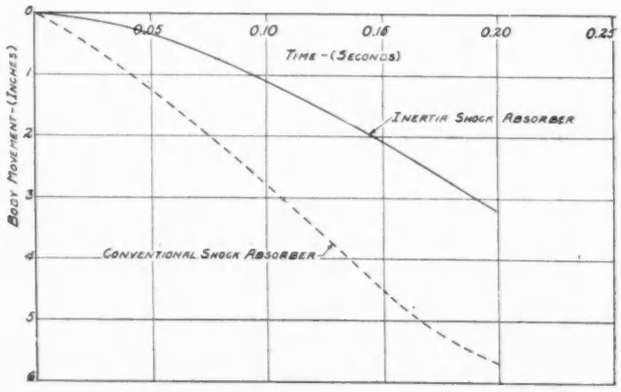


Fig. 8—Body Movement-Time Curve. Front of Car Over 4-In. Drop at 10 M.P.H.

When the tires meet an obstruction in the road, the car springs are compressed and the body begins an upward acceleration. The inertia instantly closes the valve at (*b*), so it is already closed before the car body and springs tend to separate. On the rebound side the shock-absorber is set for maximum control, and a load of several hundred pounds is immediately available to prevent the car body from accelerating further from energy stored in the spring.

In Fig. 5 is a curve showing body movements with time as controlled by the inertia absorber when the car and axles are dropped 3 in. The body and axle movements involved are similar to those produced when a car meets a depression in the road surface. The dash line represents the conventional shock-absorber and shows that the car body moves down farther than with the inertia shock-absorber. The springs are, therefore, compressed more and greater energy is stored for the rebound. The reason for this difference in action is as follows:

Differences in Action

When the wheel support is removed, the body and axle move down; but, because of the spring load, the axle moves rapidly toward the ground and separates from the body. However, the conventional shock-absorber produces resistance for this movement and tends to hold the axle toward the body; hence the wheels require a certain time before the tires touch the ground, and this time is increased with the conventional shock-absorber control. With the inertia shock-absorber, the axle moves with minimum resistance and reaches the ground quickly, at which instant spring support for the body is obtained and the body is prevented from moving as far as with the ordinary shock-absorber. After the tires touch the ground, the downward velocity of the body decreases, which is equivalent to an upward acceleration. Therefore the inertia weight closes while the car body is still moving down, and control is available the instant rebound movement begins. The amount of this control is determined by the relief valve in the shock-absorber piston; hence it is independent of temperature or viscosity and varies only slightly with velocity. In other words, with the inertia shock-absorber, maximum rebound-control is applied at the instant the body begins to move up and, although the arm velocity of the shock-absorber is a minimum, the car acceleration is a maximum and, therefore, application of the maximum control under these conditions is desirable.

The conventional orifice-controlled shock-absorber produces very little load at low velocities and consequently is not of much help in preventing the uncomfortable high accelerations that occur at the start of the rebound stroke. These comparative body movements on the rebound stroke with conventional and with inertia shock-absorbers are shown in Fig. 5. As indicated in the curves, the car body with the inertia shock-absorbers has comparatively lower values of acceleration and returns to normal position with very little disturbance to the passengers.

Fig. 6 shows the axle movement with time when the car is dropped 2 in. The curves clearly show the disadvantage inherent in the conventional shock-absorber, as the control tends to pull the car body down to a greater extent than if no shock-absorber were present. The time required for the tires to touch the ground is 0.069 sec. with conventional shock-absorbers and 0.044 sec. with inertia control.

Curves taken on body movements when the body is raised 3¾ in. by a wire and then dropped when the wire is suddenly severed are shown in Fig. 7. This demonstrates the effect of control available with inertia shock-absorbers and shows the gradual return of the car body to its normal position.

The general impression among car owners and engineers is that shock-absorbers are the most important element in determining the qualities of a ride. Although the shock-absorber must function to control its proportion of the body movements, many accelerations are unaffected by the shock-absorbers. Referring to Fig. 3, it is evident that the car springs are being compressed while the wheels are rising on the obstruction on the road surface. Assuming the axle to move 1 in. with spring rates of 350 lb. per in., there results an upward accelerating force of 700 lb. on the car frame. This force produces an undesirable acceleration of the car body, and any shock-absorber is powerless to prevent this acceleration. The fact should be borne in mind that this action occurs while the axle is rising faster than the body and exists while the tire is riding up a rise in the pavement. A similar action occurs when the tire is riding down a depression, and a decelerating force actuates the car frame downward. As previously explained, conventional shock-absorbers increase the effect of this downward force. The accelerations of the body due to these forces are affected somewhat by the mass distribution of the car but depend greatly upon the spring rates. If the spring rates were diminished by one-half, the forces and thus the body accelerations would be decreased approximately the same amount. These reactions are generally well understood but how much of riding discomfort is

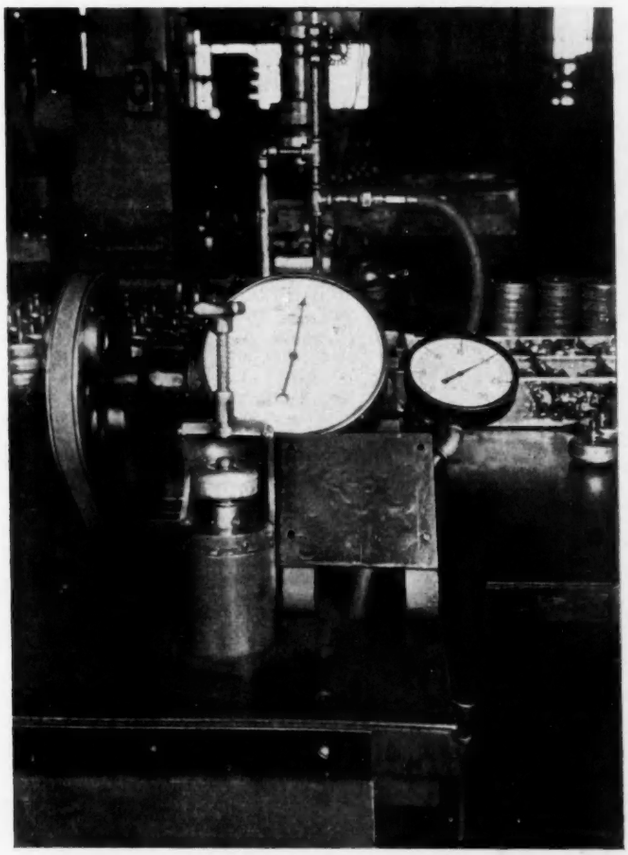


Fig. 9—Production Testing-Machine for Setting the Inertia Valve

caused by spring forces that cannot be controlled by shock-absorbers is rarely realized. The use of lower spring rates is the only answer to the problem of reducing these accelerations. The fact is realized that lower spring rates introduce problems such as axle dance, roadability and roll, and individual wheel-suspensions or other radical changes may be necessary to allow the use of light springs.

Body movements of the front end of a car driven at 10 m.p.h. along a raised track and dropped 4 in. to the floor are shown in Fig. 8. The superiority of the inertia control is plain, since the body descends at a much lower rate than with conventional absorbers. These curves were obtained by photographing a light beam as the car was driven over the drop.

The manufacture of inertia shock-absorbers required the designing of special test equipment. Usual practice was to hold the shock-absorber body stationary and move the arm to test the desired load curve. With the inertia shock-absorber, it becomes necessary to accelerate the body to determine the value of acceleration at which the valve closes.

The production testing-machine for setting the inertia valve is shown in Fig. 9. The valve-stem is moved up and down at various speeds until the valve closes. The speed is shown by a tachometer, and a change in air pressure determines the point at which the valve closes. Adjustments for the correct speed are obtained by inserting shims under the inertia-weight spring. The valve is then assembled in the shock-absorber and, after filling the body with special fluid, a test is made. The shock-absorber is placed in a machine shown in Fig. 10 which moves the body vertically while the end of the arm is attached to a spring load that registers a displacement-load diagram upon an oiled glass plate. Thus the acceleration required to close the valve is determined and, in addition, the load diagram is automatically drawn. All of our shock-absorbers pass this test and may be rejected for failing to meet the specified acceleration value or to come within the correct load limits.

Future experiments and research work undoubtedly will add to the perfection of inertia-controlled shock-absorbers. The present designs result in outstanding improvements in rides, but it is believed that further improvements in rides must come as a result of lower spring-rates and lighter wheel-suspensions.

[1] M.S.A.E.—Chief engineer, chassis spring division, Detroit Steel Products Co., Detroit.

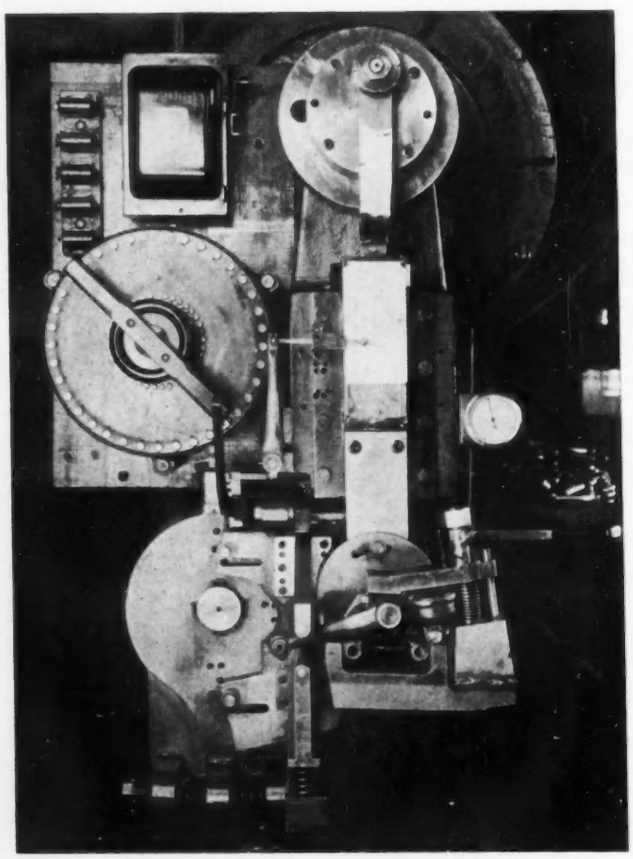


Fig. 10—Special Shock-Absorber Testing Machine

Discussion

S. P. Hess[1]*:*—It must be conceded that present-day passenger-cars do not give the public the comfortable rides which it wants. If perfect riding comfort will ever be obtained, it is in the future; many factors enter into its final solution, such as tires, springs, seat cushions and shock-absorbers. Their functioning must harmonize at all speeds, with varying passenger loads, over smooth or rough roads. Every car engineer has some degree of experience in riding qualities and is familiar with the variables which make up the problem and influence its ultimate solution.

I believe that much can be done with a modification of the conventional spring suspension. By this I mean the method of attaching the springs to the car, rather than in a radical change in the design of the springs. All conditions being considered, the spring engineer can obtain any flexibility or rate desired if physical dimensions of the spring will permit this with safety. Wider spring centers or outboard spring mountings will add steadiness to the car by eliminating sidesway, and at the same time give a good fore-and-aft ride.

It seems strange that, after all the discussion on riding comfort, so few car engineers have the same version of a good ride, or will commit themselves definitely on the subject. One engineer may insist upon stability at high speed, which means stiff springs and heavy shock-absorber adjustments. Another insists on a soft "boulevard" ride, which means soft springs and light shock-absorber adjustment. The public must give the final answer.

Unfortunately, many steps taken to improve riding in recent years developed other troubles such as shimmying, tramping, wheel-fight and the like. Most of these were corrected, although many were at the expense of a compromised ride.

Motor Vehicles Help Railroads

"THE railroads are an absolute necessity to this country and when the smoke has cleared from the scene the railroads will find that trucks and buses are just as essential to them as they are to the balance of the community, and that rather than being a detriment to passenger and freight operation, they are contributors."—A. F. Coleman, manager, Motor Vehicle Dept., Standard Oil Co., of New York.

Is 50 Miles Per Gallon Possible With Correct Streamlining?

By W. E. Lay

THIS is Part 2 of a study of air resistance in terms that the automobile engineer can understand without delving deeply into aerodynamics. In Part 1, after analyzing car resistance mathematically, the author related how air resistance was determined by wind-tunnel tests of various body models and presented tabulated and charted results. A study of the test methods used is presented herewith, together with comparisons made between the results obtained in Part 1 and those obtained in Part 2 from road tests of a car equipped with a so-called "floating envelope."

Fuel consumption is considered also, since full advantage of streamlining cannot be obtained without improvement of the transmission to provide for sufficient activity of a car at the lower speeds.

In conclusion, the salient facts of the entire paper are summarized and seven specific suggestions for streamlining are made to car builders.

Part 2

THIS investigation has been not only a study of models, as published in Part 1, but of test methods. To determine the drag of an airfoil or balloon model is comparatively simple, whereas the test of an automobile model is complicated by the necessity for providing a road surface. We find that, 22 years ago, Eiffel tried to use a running belt and found it impracticable because of an uncontrollable flapping. An apparatus capable of driving a moving belt at 90 m.p.h. would be so large that, unless it were built integral with the tunnel, it would produce greater disturbances in the airstream than would the model itself. Ten years ago Jaray used a flat plate as a road surface, but we know that such an arrangement will not duplicate the conditions under which a car actually operates. It can be built and placed in the airstream in such a way that it will create little or no disturbance of the air flow in the region about the model except that it has a boundary layer extending up toward the model. A layer of air is attached to the plate surface and does not move at all. Succeeding layers above the plate flow at increasing speeds until, at 0.60 in. above the plate, the velocity is only 2 or 3 per cent less than that of the main airstream. There is, moreover, a similar boundary layer on the under surface of the model itself and the two act to cause a much reduced velocity of the air moving underneath the body.

It seems reasonable that, if the boundary layer could be removed from the plate surface so that the air flow along the plate surface is not retarded, the model would be surrounded by the same conditions as the actual car. This suggestion was made by Professor Stalker, and such a plate was made with some 2000 holes drilled in its upper surface, through which the air of the boundary layer was removed by suction.

The remaining method, called the aerodynamic reflection method, used by Rumpler, Pawlowski and others, consists of placing two identical models bottom to bottom. This will produce between them a neutral plane of air flow whose maximum velocity may be practically that of the main airstream, while the pressure effects on the under surface of the model tested should approach those of the car operating on the road. The lower model is supported entirely by guy wires and only the upper model is tested.

The tests were conducted in a wind-tunnel with an Eiffel chamber and double return ducts, as shown in Fig. 20. The octagonal throat was reduced to 6-ft. diameter to obtain speeds up to 90 m.p.h. Test runs were made at speeds from 30 to 90 m.p.h. The method of supporting the models in the airstream, as shown in Fig. 21, made it possible to measure not only the drag but also the front and rear lift and the pitching moment. Tests were made also with the models hanging free, without provision for the road surface, as shown

[This paper was presented at the 1933 Annual Meeting of the Society. Part 1 was published in the S.A.E. JOURNAL, April, 1933, p. 144. The author is a Member of the Society and professor of mechanical engineering at the University of Michigan, Ann Arbor, Mich.]

Table 11—Summary

Description	Model No.	K	Reduction, Per Cent	Factor
Flat Plate	0	0.00327	....	15.57
Box, Sharp Corners	1	0.00216	....	10.30
Box, Rounded Corners, 6-in. radius	2	0.00151	....	7.20
Box, Rounded Corners, 9-in. radius	2a	0.00113	....	5.37
Box, R.C., Wheels Outside	3	0.00135	....	6.43
Box, R.C., Wheel Pockets Closed	4	0.00148	....	7.05
Box, R.C., Wheel Pockets Open	5	0.00146	....	6.95
Bulging Model, 45-Deg. x 45-Deg.	6	0.000409	....	1.95
Vertical Windshield, Stub Rear	7	0.000886	0.0	4.21
Vertical Windshield, Medium Rear	8	0.000880	0.7	4.18
Vertical Windshield, 45-Deg. Rear	9	0.000825	6.9	3.93
Vertical Windshield, Streamline Rear	10	0.000616	30.4	2.93
22½-Deg. Windshield, Stub Rear	11	0.000799	9.8	3.80
22½-Deg. Windshield, Medium Rear	12	0.000661	25.3	3.12
22½-Deg. Windshield, 45-Deg. Rear	13	0.000642	27.5	3.05
22½-Deg. Windshield, Streamline Rear	14	0.000442	50.1	2.10
45-Deg. Windshield, Stub Rear	15	0.000758	14.4	3.61
45-Deg. Windshield, Medium Rear	16	0.000579	34.6	3.75
45-Deg. Windsheild, 45-Deg. Rear	17	0.000536	39.4	2.55
45-Deg. Windshield, Streamline Rear	18	0.000309	65.1	1.47
Elliptical Front, Stub Rear	19	0.000763	13.9	3.63
Elliptical Front, Medium Rear	20	0.000606	31.6	2.89
Elliptical Front, 45-Deg. Rear	21	0.000504	43.1	2.40
Elliptical Front, Streamline Rear	22	0.000320	63.9	1.52
Practical Streamline Model	23	0.000651	26.4	3.10
Practical Streamline Model, Flat Radiator	24	0.000825	6.9	3.93
Practical Streamline Form V-45-Deg. Radiator	25	0.000732	17.3	3.48
Practical Streamline Form 105-Deg. V-Vert. Windshield	26	0.000825	6.9	3.93
Practical Streamline Form 105-Deg. V-45-Deg. Windshield	27	0.000770	13.0	3.67
Practical Compromise Car Model	28	0.000802	9.4	3.82
1930 Sedan Model	29	0.001547	....	7.37
Ideal Streamline Form	100	0.000210	76.3	1.00

in Fig. 22. In Fig. 23 is shown the plate which was used first without removal of the boundary layer according to Eiffel's method. It was used also when the boundary layer was partly removed according to Stalker's suggestion. In Fig. 24 is shown the set-up of models by the reflection method, the lower model being supported by guy wires and in no way connected to the upper model or to the weighing scales. Results are summarized in Table 11.

All of the wind-tunnel-test results are given in terms of K, the drag coefficient, expressed in pounds per square foot of area. Although not exactly correct, it was convenient to work out the results assuming the drag to vary as the square of the velocity. All data were plotted on logarithmic graph paper and a straight line faired through the experimental points. The error involved in this method was within the limits of experimental error over the velocity range covered. These results also were plotted against speed. Fig. 25 shows such drag curves for the rectangular models with sharp and with rounded corners and edges. These particular results show an apparent change in air-flow conditions occurring at 80 m.p.h. for the model with rounded edges of 6-in. radius, and at 60 m.p.h. for the models with 9-in. radius edges. This change was not encountered in any of the other methods of test.

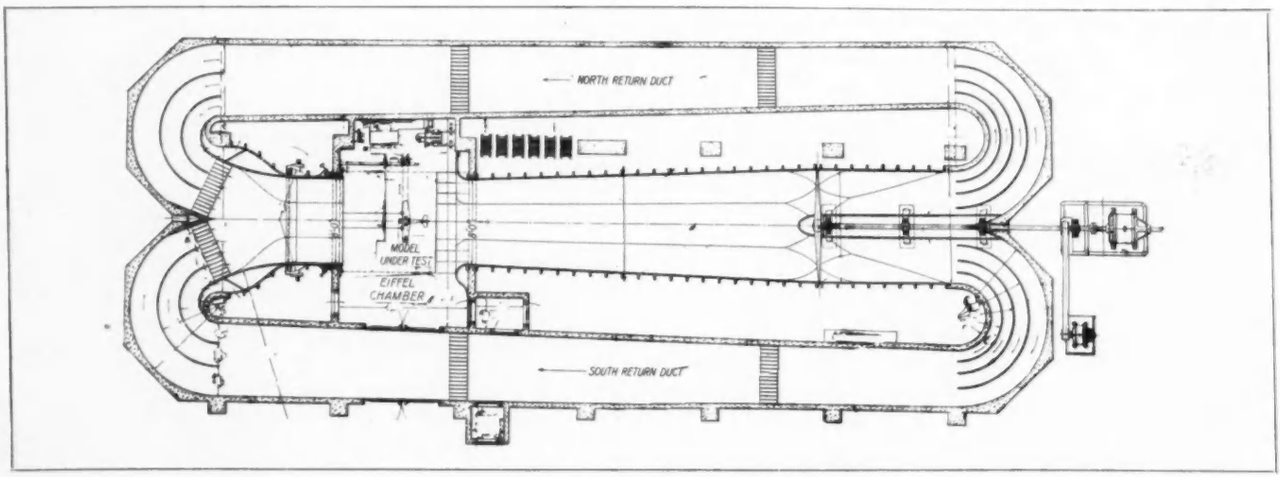


Fig. 20—University of Michigan Wind Tunnel

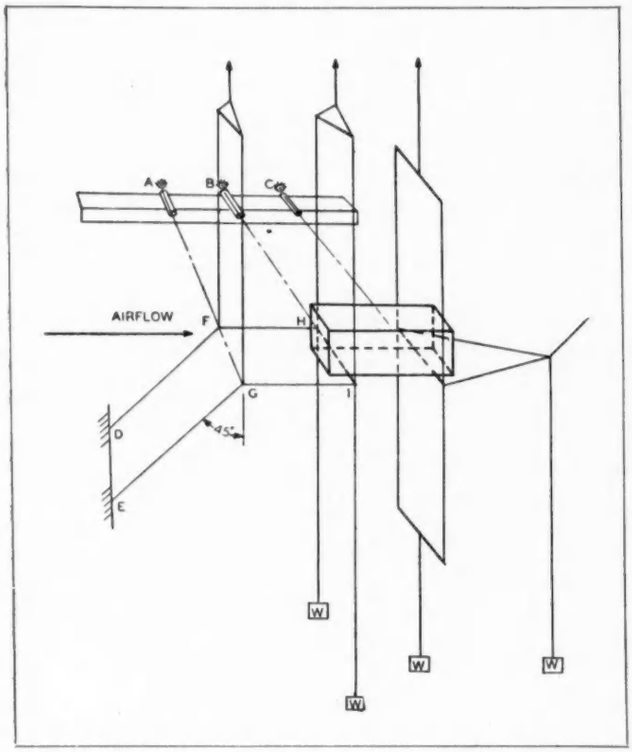


Fig. 21—Wind Tunnel Set-Up of Model

Tests were made on Model No. 28 by the five different methods listed below and the results obtained are compared in Fig. 26 and in Table 12.

Method *A*—using the flat plate without removal of boundary layer

Method *B*—using the plate with partial removal of the boundary layer by applying one-half the maximum suction

Method *C*—using the plate and applying full suction to remove about one-third of the boundary layer

Method *D*—hanging the model free in the airstream without making provision for the road surface

Method *E*—using the reflection method with two identical models

This indicates clearly the manner in which the boundary layer on the testing plate affects the measured resistance of the model. With full suction applied to the 2000 holes in the plate, only one-third of the boundary layer was removed. Extrapolation of the results obtained with the plate indicates that, if the boundary layer could be completely removed, the test results would agree with those obtained by the reflection method. It is better to hang the model free in the test chamber without any provision for the road surface than to use a flat plate without complete removal of the boundary layer. We believe that the reflection method is the best to use, despite the fact that it required duplication of all body models.

Such wide variation in the results obtained by different methods of wind-tunnel test made it quite evident that we were still uncertain as to the actual air resistance of these models when built to full scale and operated on the road. By building some of these models to full scale and testing them on the road, we would be able to establish the relationship between the results obtained by some type of wind-tunnel test of a one-eighth scale model and the actual air resistance of that body as it was driven along the road. Having admitted failure to duplicate the effect of the road in the wind-tunnel, we may yet establish a factor by which the value of K, obtained by wind-tunnel test, may be multiplied to give the value of K to be applied to the car as it operates on the road.

In making a survey of the practical methods of determining the air resistance of full-size motor-vehicles under actual operating conditions, we considered possibilities: (*a*) the towing method; (*b*) deceleration on a level road; and (*c*) coasting down a uniform grade. It will be noted that in all three of these methods the total resistance is first determined. Then, by towing or test on a chassis dynamometer, the rolling resistance is determined and subtracted from the total to give the air resistance.

$$R_a = R - R_r \tag{8}$$

where

R_a = air resistance
R = total resistance
R_r = rolling resistance

The first method (*a*) consists of using one car to tow another at the end of a long cable. On one end of the cable is attached a recording drawbar-pull dynamometer which records the force transmitted through the cable from the towing to the towed car. This method had been given a thorough trial in 1931 and discarded because of the terrific turbulence set up in the wake of the towing car. Method (*b*) consists of driving the car up to its top speed and then allowing it to coast in neutral to a stop, noting the time required for say each 5-m.p.h. decrease in speed with some type of recording chronometer. From these data the deceleration, or rate of speed change, can be computed for each speed.

If a car is coasting or decelerating on a level road, not only is its speed decreasing but its stored kinetic energy is decreasing. This decrease in energy gives rise to a force that is applied to drive the car forward against the rolling resistance and the air resistance, as shown in Fig. 27.

$$F = M_a = (W_a/g) = R_r + R_a \tag{9}$$

$$K = [W_a/g) - R_r] \,/\, AV^2 \tag{10}$$

where

W = gross weight of the vehicle, in pounds
g = acceleration due to gravity, in feet per second per second
a = rate of deceleration for the vehicle, in feet per second per second
R_r = rolling resistance per second
A = projected area of the vehicle, in square feet
V = vehicle speed, in miles per hour

Table 12—Methods of Wind-Tunnel Test

Description	Model No.	K	Variation, Per Cent
A—Using Flat Plate, no suction	28	0.00594	74.0
B—Using Flat Plate, half suction	28	0.00621	77.3
C—Using Flat Plate, full suction	28	0.00685	85.2
D—Model Free, like airship test	28	0.00761	94.5
E—Reflection Method, using two models	28	0.00803	100.0

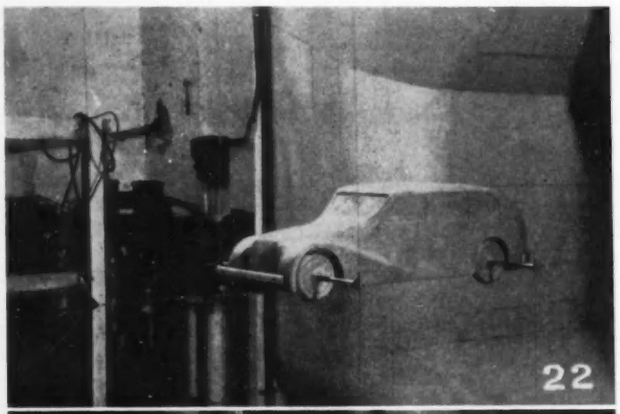
Equation (10) assumes, however, that all of the kinetic energy stored in the vehicle is that due to its linear motion. Additional energy is stored in parts which have not only linear but also rotary motion, so the total energy is

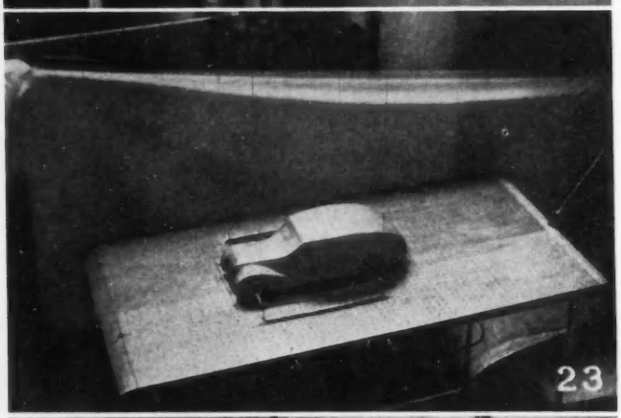
$$\text{Kinetic Energy} = \tfrac{1}{2}\,(W/g)\,v^2 = I\,\omega^2 \qquad (11)$$

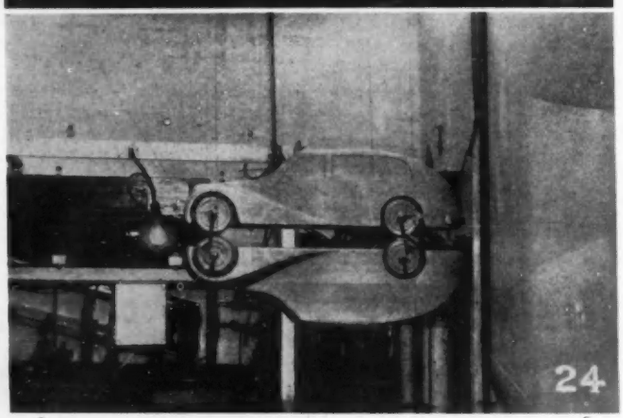
where

v = vehicle velocity, in feet per second
I = moment of inertia of the rotating parts
ω = angular velocity of the rotating parts, in radians per second

It is therefore necessary to determine the moment of inertia of all rotating parts, which include: (*a*) all wheels, with their tires; (*b*) rear-axle shaft, with differential casing and gears; (*c*) propeller-shaft, with universal-joints and pinion; and (*d*) transmission main shaft, with its gears.







Wind-Tunnel Test of Models

Fig. 22—Model hanging free. **Fig. 23**—Model used with a plate. **Fig. 24**—Model being tested by the "reflection method."

After the moment of inertia of all rotating parts has been determined, it is more convenient to calculate an equivalent weight whose kinetic energy due to the linear speed of the car would be equal to the kinetic energy in all of the rotating parts when the car is moving at that speed.

$$K = \{\,[(W + w_a)/g) - R_r]\,\}\,/\,AV^2 \qquad (12)$$

where w = the equivalent weight whose kinetic energy at any car speed is equal to the kinetic energy due to rotation of parts. The moment of inertia of the wheels and tires may be easily obtained but that of the other three items is determined with great difficulty.

The coasting method consists of driving a car over the brow of a hill of uniform grade and coasting down the grade at uniform speed. The speed maintained at a constant value by any car on a given grade is determined by trial. Once that speed is determined, we know that at that speed the component of the vehicle weight which tends to roll the vehicle down hill is equal to the rolling resistance plus the air resistance, as shown in diagram in Fig. 28.

$$F = R_r + R_a = W \sin \alpha \qquad (13)$$

$$\tan \alpha = a/b = \sin \alpha \text{ for grades less than 8 per cent without serious error} \qquad (14)$$

$$F = W\,(a/b) \qquad (15)$$

$$(a/b) \times 100 = G \qquad (16)$$

$$K = [\,(WG/100) - R_r]/AV^2 \qquad (17)$$

where G = grade, in per cent, and α = angle of grade, in degrees. If a determination is to be made at any other speed, a hill of a different uniform grade must be located and new tests made.

Just as we had decided to adopt the deceleration method, there came an inspiration; we would build a "floating envelope." From a low-priced car which had been loaned to us, we removed the doors, hood, fenders and bumpers. About the regular coupé body was built the floating envelope, corresponding in shape to the rectangular box tested in the wind-tunnel. This envelope, shown in Figs. 29 and 30, was made of pressed-wood panels fastened to a light wooden frame well braced with wood braces and piano wire fitted with airplane-type turnbuckles. The corners and edges are detachable, so that sharp edges and rounded edges of 6 and 9-in. radius can be investigated. The edges are made of wire mesh fastened over a light framework, with airplane fabric stretched over the whole and doped so that it is as taut as a drum-head. All joints are covered with doped fabric.

On the front and rear bumper-supports are mounted four ball-bearing rollers, shown in Fig. 31. The framework of the envelope is supported by steel tubing which can move forward or backward on the grooved rollers. Thus the envelope is free to move forward or backward several inches on the ball-bearing rollers. As the car inside the envelope is driven forward, the air pressure tends to force the envelope to the rear. This motion is restrained only by a piano wire connecting the envelope to a drawbar dynamometer, mounted in the rear of the car as shown in Fig. 32. This wire transmits from the car inside to the envelope the force required to drive the envelope forward through the air. The forward end of the wire is attached by a link to a knife-edge on the short arm of a bell-crank, which in turn pulls down on a Chatillon spring-scale.

Table 13—Air Resistance of Full-Scale Body

(Rectangular Box with 9-In. Radii)

Speed M.P.H.	Measured Air Resistance, Lb.	K in $R_a = KAV^2$
30	30.0	0.00104
40	53.4	0.00104
50	81.5	0.00102
60	120.0	0.00104
70	163.0	0.00104
	Average	0.00104

Table 14—Results of Wind-Tunnel Tests of Models with and without Wheels

Model	K
A—Rectangular box; sharp corners and edges, with wheels	0.00220
B—Rectangular box; sharp corners and edges, without wheels	0.00219
C—Rectangular box; 6-in.-radius corners and edges, with wheels	0.00151
D—Rectangular box; 6-in.-radius corners and edges, without wheels	0.00146
E—Rectangular box; 9-in.-radius corners and edges, with wheels	0.00118
F—Rectangular box; 9-in.-radius corners and edges, without wheels	0.00113

Table 15—Air-Resistance Coefficient Determined by Wind-Tunnel and by Road Test

(Rectangular-Box Model, 9-in.-Radii Corners and Edges, without Wheels)

	K
Determined by Wind-Tunnel Test	0.00113
Determined by Road Test	0.00104

$$\text{Factor} = \frac{0.00104}{0.00113} = 0.92$$

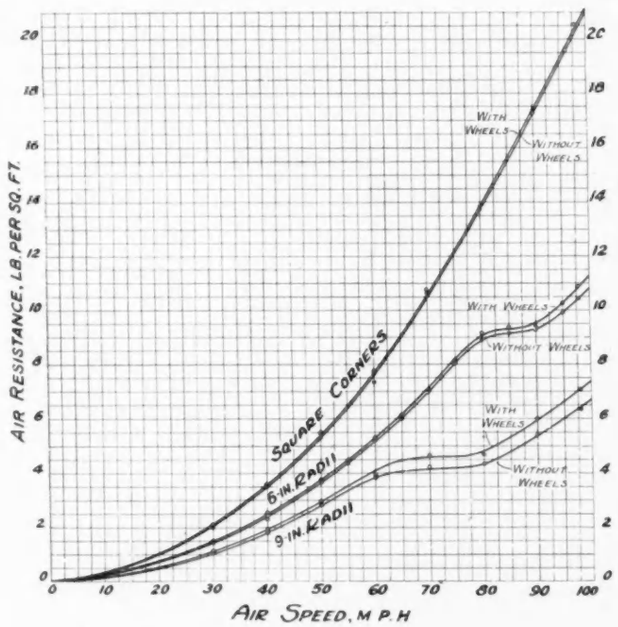


Fig. 25—Air Resistance of Rectangular-Box Models by Wind-Tunnel Tests

Drag curves, both with and without wheels, are shown. They illustrate the remarkable reduction obtained by round corners and edges.

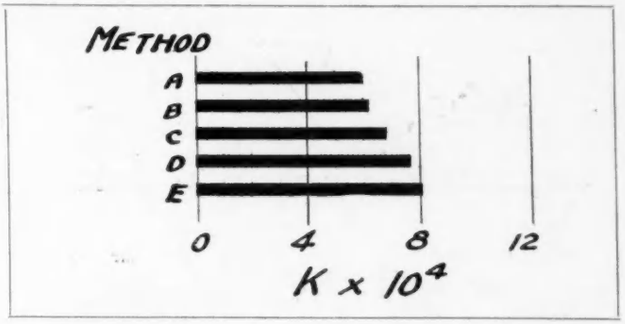


Fig. 26—Comparison of Results Obtained by Different Test Methods

A four-mile course of approximately level, smooth concrete road was located. A mile of this road in the middle of the course was accurately measured. Red lights were placed at each end of the measured mile, and at one end of the course a miniature Weather Bureau station was set up to measure wind velocity and direction and the temperature, pressure and humidity of the air. The car was driven over the full length of the course on each run. On starting the run, the throttle was adjusted at once to a point found by previous trial to give approximately the desired speed. At the beginning of the test mile, two stop-watches were started and the observer began taking readings on the dynamometer as rapidly as possible. At the end of the test mile, both watches were stopped and the average speed over the course computed. Immediately thereafter a test run was made in the opposite direction. This was continued until wind conditions made further test runs impossible.

The dynamometer was calibrated before and after each set of runs to make certain that no part of the apparatus had shifted. From the calibration results the scale readings made on the test were converted into air-resistance force in pounds. These results were then corrected to apply to standard air conditions; 59 deg. fahr. temperature, 29.92-in. mercury pressure, and density of 0.076511 lb. per cu. ft., assuming that the air resistance varies directly with the density of the air.

$$\text{Density of Air} = \frac{0.080723}{1 + 0.002039\ (t - 32)} \cdot \frac{B - 0.378e}{29.92}$$

where

t = dry-bulb temperature, in degrees fahrenheit
B = observed barometric pressure corrected to 32 deg. fahr.
e = water-vapor pressure, in inches of mercury, as obtained by Carrier's equation

The values of the air resistance are then plotted on rectangular coordinates against car speed, as shown in Fig. 33. The average of a group of points is obtained by computing the mean force for the mean speed, assuming the force to vary as the square of the speed. When a smooth curve has been drawn, data read from the curve give us the resistances and coefficients stated in Table 13.

Through the kind cooperation of the staff at the General Motors Proving Ground, tests were made on the full-scale body by the deceleration method. The average value of K over a speed range of 15 to 55 m.p.h. was found to be 0.00103, which is in excellent agreement with the results in Table 13.

In the road tests made by the floating-envelope method, the road wheels are driven through the air but the air resistance is not included in our measurements. This, together with a slight change in the wheelbase, required that new

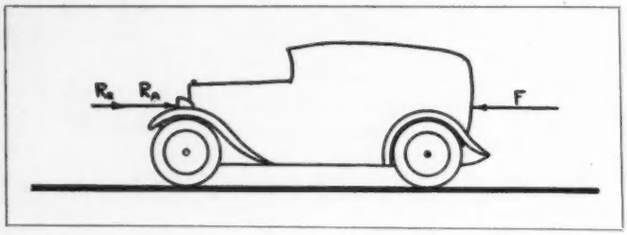


Fig. 27—Diagram of Forces Acting on a Car When Coasting on a Level Road

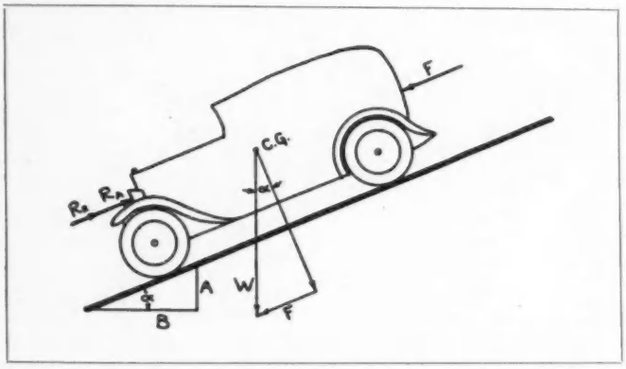


Fig. 28—Diagram of Forces Acting on a Car When Coasting Down a Grade at Constant Speed

models be made and tested with the wheels in place but supported on the reflection model. Tests were made also with the wheels fastened to the model under test, with results shown in Table 14.

We are now in position to compare the wind-tunnel-test results made on a one-eighth scale model with road-test results on a full-scale model, as in Table 15.

Such are the results so far, meager indeed but indicative of a research problem fully worthy of our mettle. Probably this is the first time that the air resistance of a motor-vehicle has been determined directly on the road under actual operating conditions. The work has been surrounded with many difficulties and only by tedious trial and error has the technique of test been developed.

Better Fuel Mileage with the Gear Ratio

Full advantage of streamlining cannot be obtained without improvement of the transmission to provide for sufficient activity of the car at the lower speeds. Moreover, the fuel mileage can be greatly increased by the correct choice of gear ratios. The combination of the final-drive ratio with the three or four forward speeds in the gearset is necessarily a compromise. Using the conventional design, we are obliged to use a gear ratio, chosen to give grade acceleration, under conditions where a high fuel-mileage is most desired. To understand clearly what can be accomplished, the fuel consumption of the engine must be known, from idle to full throttle, for every speed over the operating range of the engine.

In Fig. 34 are shown the performance curves for a typical small-car engine. Its brake horsepower, torque and specific fuel consumption, stated in terms of pounds per brake horsepower hour, are plotted against speed. The fuel consumption is given for each speed, not only at full throttle, but at ¾, ½ and ¼-load operation. At a given percentage of engine output, the fuel consumption is low in the middle range, somewhat higher at low speeds and especially high at high speeds. The same data may be plotted on the basis of percentage of full load, with a separate curve for each speed, as shown in Fig. 35. With this arrangement of the data, it is clearly evident that at any given speed the fuel required to develop 1 hp. for 1 hr. increases very rapidly as the power output is reduced by closing the throttle.

Table 16—Performance with Variable Gear Ratio (Normal Gear Ratio, 3.25)

For Hill-Climbing or Acceleration

1 Car Speed, M.P.H.	2 Engine Speed, R.P.M.	3 Maximum Engine Power	4 Maximum Available Power	5 Gear Ratio
20	3,250	58.8	50	13.80
30	3,250	58.8	50	9.22
40	3,250	58.8	50	6.91
50	3.250	58.8	50	5.53
60	3.250	58.8	50	4.61
70	3,250	58.8	50	3.95
80	3,250	58.8	50	3.46
85	3,250	58.8	50	3.25

For Level-Road Operation

1 Car Speed, M.P.H.	2 Engine Speed Normal Ratio, R.P.M.	3 Total Resisting Power	4 Engine Power	5 Maximum Engine Power	6 R.P.M. of Maximum Power	7 Desired Engine Speed	8 Per Cent of Maximum Engine Load	9 Factor	10 Desired Gear Ratio	11 Specific Fuel Consumption, Lb. per B.Hp.	12 Fuel Consumption Lb. per Hr.	13 Fuel Consumption Gal. per Hr.	14 Fuel Mileage, Miles per Gal.
20	765	3.0	3.53	4.17	100	600	27.2	0.392	1.28	1.55	5.47	0.875	22.9
30	1,150	5.3	6.25	7.35	240	600	48.0	0.392	1.28	1.12	7.00	1.120	27.8
40	1,530	9.0	10.60	12.50	550	600	81.5	0.392	1.28	0.85	9.00	1.440	27.8
50	1,910	14.0	16.50	19.50	890	890	85.0	0.465	1.51	0.74	12.20	1.950	25.6
60	2,295	21.0	24.70	29.20	1,320	1,320	85.0	0.574	1.87	0.67	16.50	2.640	22.7
70	2,680	30.1	35.40	41.80	1,960	1,960	85.0	0.730	2.37	0.65	23.00	3.680	19.0
80	3,060	42.4	50.00	58.00	3,060	3,060	85.0	1.000	3.25	0.78	39.00	6.230	12.8
85	3,250	50.0	58.80	58.80	3,250	3,250	100.0	1.000	3.25	0.80	47.10	7.550	11.2

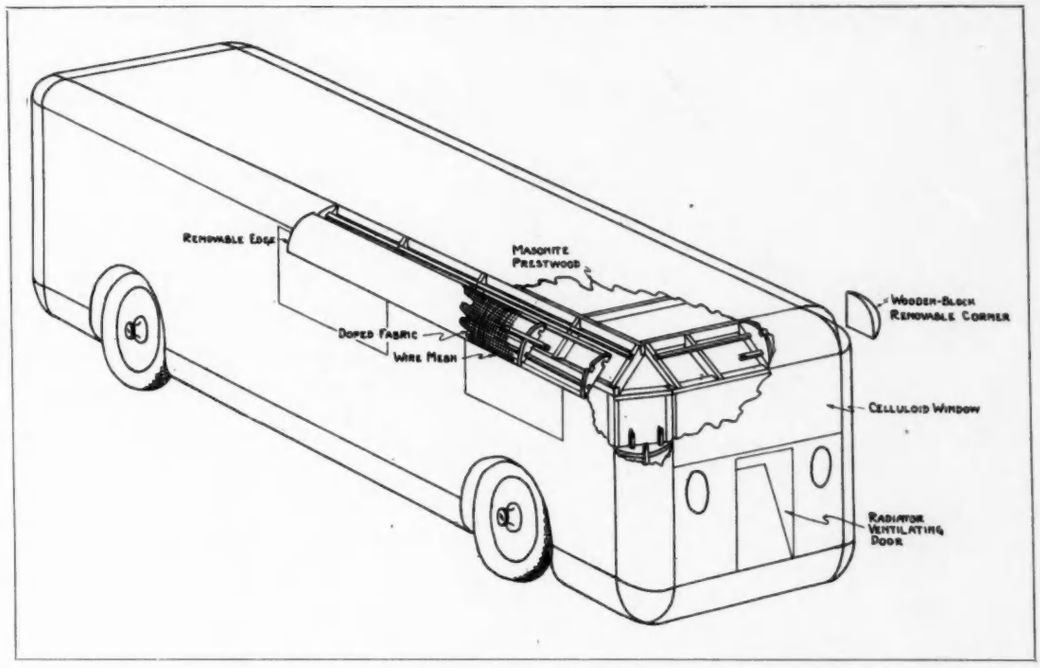


Fig. 29—Sectional Diagram of the "Floating Envelope"

Table 17—Performance with Typical Gear Ratio (High Gear Ratio 4.60)

For Hill-Climbing or Acceleration				For Level-Road Operation							
1	2	3	4	1	2	3	4	5	6	7	8
Car Speed, M.P.H.	Engine Speed, R.P.M.	Maximum Engine Power	Maximum Available Power	Car Speed M.P.H.	Total Resisting Power	Engine Power	Per Cent of Maximum Power	Specific Fuel Consumption, Lb. per B. HP.	Fuel Consumption Lb. per Hr.	Fuel Consumption Gal. per Hr.	Fuel Mileage, Miles per Gal.
20	1,083	23.8	20.4	20	3.5	4.12	17.3	1.67	6.9	1.10	18.2
30	1,626	35.1	30.0	30	7.0	8.25	23.5	1.23	10.2	1.63	18.4
40	2,165	45.1	38.3	40	13.1	15.4	34.1	1.03	15.9	2.54	15.7
50	2,710	54.0	45.9	50	22.0	25.9	48.0	0.92	23.8	3.81	13.1
60	3,250	58.8	50.0	60	34.9	41.1	70.0	0.90	37.0	5.92	10.1
67	3,640	55.2	47.0	67	47.0	55.4	100.0	0.98	54.2	8.68	7.7

Let us next determine what engine power is required under different conditions. Plotted in heavy line against speed in Fig. 36 is shown the total resisting power of the car in which the above engine was mounted. On this plot there is also found the power available at the rear wheels to drive the car in each of the three forward speeds. The top speed of this car is reached at 67 m.p.h. Let us suppose that by streamlining the body we can reduce the air resistance by one-half. The intersection of the available-power and the resisting-power curve will then indicate a top speed of 74 m.p.h. Should the rear-axle ratio be changed so that the intersection comes at the peak of the available-power curve for high gear, the top speed of the car will be increased to 85 m.p.h. Let us call this last gear ratio the normal gear ratio.

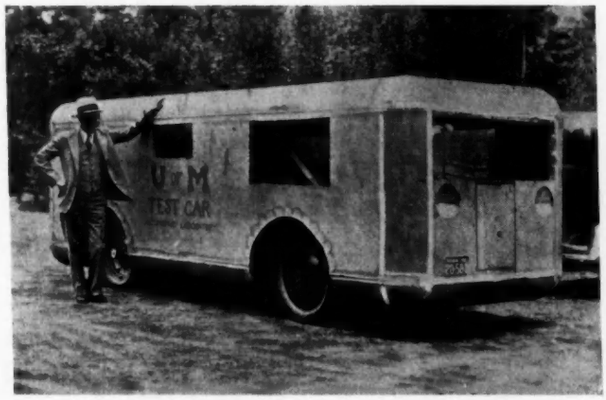


Fig. 30—Car Equipped with a "Floating Envelope"

To simplify the computation and analysis, the efficiency of the transmission system, including everything from clutch to rear wheels, is assumed to be 85 per cent under all operating conditions. With this assumption, shifting from high to second or to low gear does not change the quantity of available power but merely reduces the speed range over which it is available. At any of the lower speeds the available power in excess of that needed to move the car on the level road is greatly increased by shifting to a lower gear ratio. At 20 m.p.h. there is available for acceleration or hill climbing 17 hp. in high, 27.3 hp. in second, and 45 hp. in low gear. To obtain the best acceleration and the best hill-climbing ability at any speed, a gear ratio should be chosen that will move the peak of the available-power curve over to that car

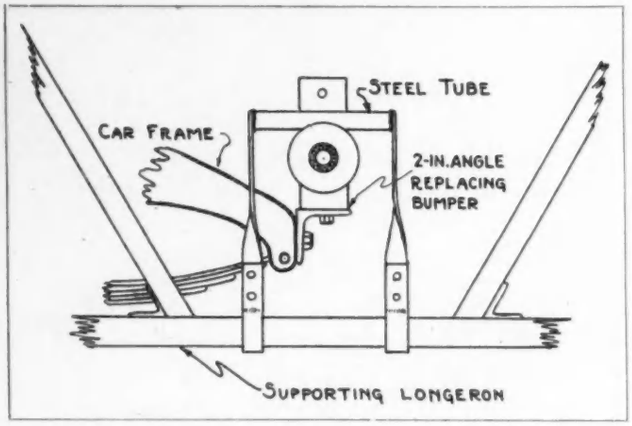


Fig. 31—Ball-Bearing Roller Allowing Longeron and Floating Body to Move Fore and Aft

speed. This suggests at once a variable speed transmission in which the gear ratio may be changed by small increments over a fairly wide range. With such a transmission the power available at any speed is the maximum, as shown by the horizontal dash line at the top in Fig. 36.

Optimum Ratio

Assuming that such a transmission were available, what is the optimum ratio for each speed and condition of operation?

(1) *For Best-Car Activity.*—For each car speed the gear ratio chosen should allow the engine to operate at the peak of its power curve.

(2) *For Level-Road Operation To Give Best Fuel Mileage.*—For each car speed the gear ratio chosen should cause the engine to produce the required power at low engine-speed and at nearly full-open throttle.

Let us consider what could be done with this transmission in a car streamlined to give one-half the air resistance of the typical car herein discussed. From Fig. 36 it is seen that 21 hp. at the rear wheels is required to drive the car on the level road at 60 m.p.h. The engine must then develop $21.0/0.85 = 27.4$ hp.

Under what operating conditions will the engine develop this power, using the least fuel? The answer is, at low speed and open throttle. Reducing the gear ratio will bring about both of these results. Because of roughness in operation and the use of a richer mixture at full throttle, it may be best to operate the engine at, say, 85 per cent of its full-throttle power. The maximum power at the desired engine speed is then $24.7/0.85 = 29.2$ hp. In Fig. 34 this corresponds to an engine speed of 1320 r.p.m. Dividing this number by the engine speed when using the normal gear-ratio at 60 m.p.h., we obtain a factor $13.20/2295 = 0.575$ by which the normal gear-ratio is multiplied to obtain $3.25 \times 0.575 = 1.87$, the desired over-all gear-ratio. At low car-speeds the engine speed computed in this manner becomes too low for good operation, so that below 40 m.p.h. the engine speed may be kept at 600 r.p.m. and the throttle closed as necessary to obtain the lower speeds. As the top speed is approached the throttle is, of course, fully opened.

Fuel Mileage Computed

We may compute the fuel mileage as follows: At 60 m.p.h. the engine is operating at 85 per cent of full power at 1320

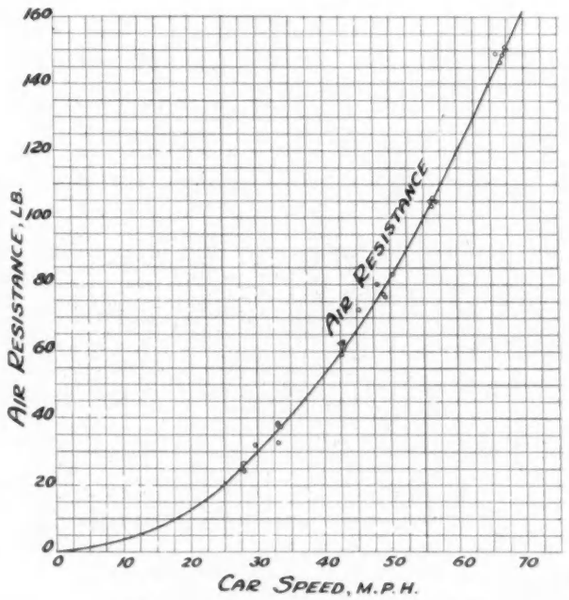


Fig. 33—Drag Curve of the "Floating Envelope"

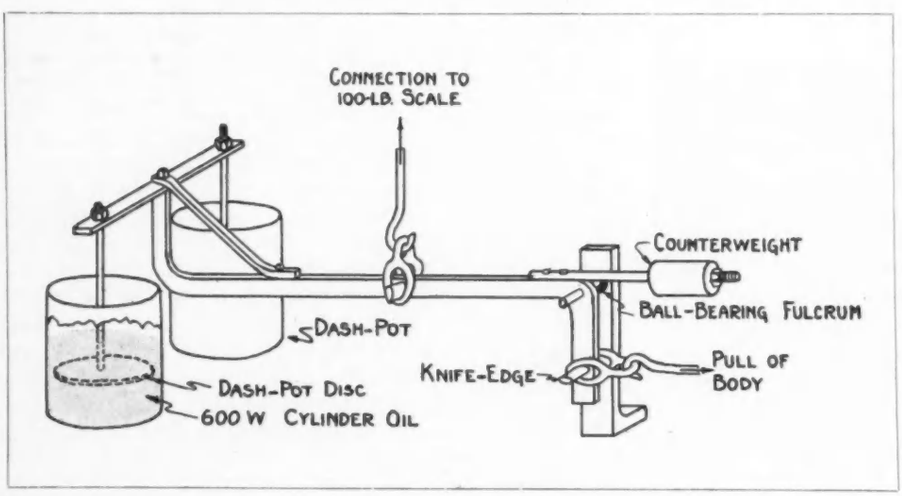


Fig. 32—Drawbar Dynamometer for Measuring Air Force on the "Floating Envelope"

r.p.m. In Fig. 34 its specific fuel consumption is then 0.67 lb. per b.hp. The engine is using $(0.67 \times 24.7)/6.25 = 2.64$ gal. per hr. while it travels a distance of 60 miles with a fuel mileage of $60/2.64 = 22.7$ miles per gal.

In Table 16 are given the computations for speeds of 20 to 85 m.p.h. on the level road and also the conditions for maximum car activity. In Table 17, similar data are arranged for the conventional car.

In Fig. 37 the level-road fuel-mileage to be obtained by changing gear ratios in this manner is plotted against speed, where it may be compared with a similar plot for the conventional car using the same engine. Over the normal driving range for our streamlined car, say from 50 to 65 m.p.h., the fuel mileage is increased from 100 to 150 per cent.

Specific Fuel Consumption Plotted

The specific fuel consumption of the engine as it operates at each car speed may now be plotted on Fig. 34 by dash lines, as shown, for both the streamlined car with the variable transmission and the conventional car. The number at each point indicates the car speed in miles per hour. This shows clearly how the engine operates with much lower fuel consumption using the variable-gear-ratio scheme.

In Fig. 38 is plotted against speed the desirable gear-ratio at each speed for both maximum fuel mileage on the level road and for maximum car activity. It will be noted that this latter gear ratio approaches infinity at the lowest speed. Such large gear-ratios will spin the wheels on any road surface, so a limit like that shown should be placed on the gear-ratio available.

Such a variable-speed transmission will not only provide the maximum car activity at all speeds but greatly increase the fuel mileage attained by the car. The greatest improvement in fuel mileage is attained in the normal driving range, where it will be most appreciated by the average owner. If this transmission-gear ratio can be shifted while transmitting power, the improvement in car acceleration will be remarkable.

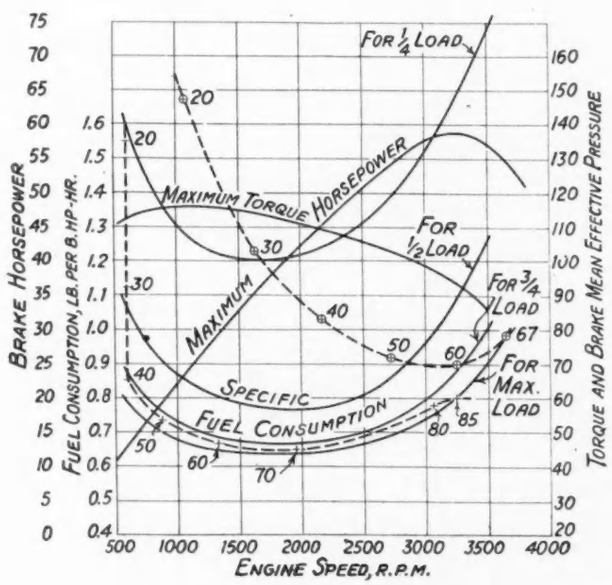


Fig. 34—Engine Performance Curves

The curves show the fuel consumption at ¾, ½ and ¼ load, as well as that at maximum power output.

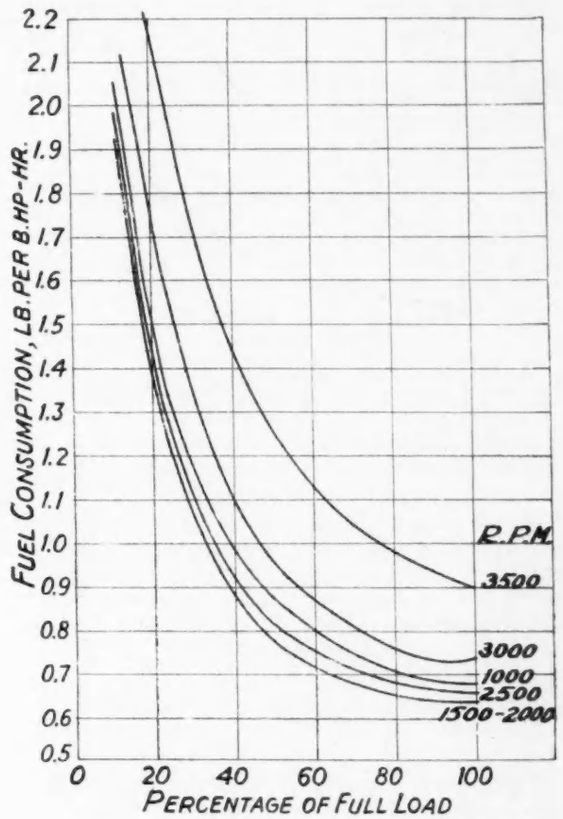


Fig. 35—Fuel Consumption at Constant Engine Speeds

Summary

The salient facts presented in this paper may then be summarized by the following brief statements:

(1) That determination of the air resistance of a motor-vehicle is complicated by the fact that it seldom moves straight into the air body and that it operates close to the ground.

(2) We are not certain that we can reproduce in the wind tunnel the effect of the road surface.

(3) We have been able to measure directly the air resistance of a full-scale body by the "floating envelope" method.

(4) We have shown that the normal driving speed is increasing.

(5) Maximum car speeds are increasing.

(6) The increase has been accomplished mainly by the use of larger engines.

(7) This policy if continued will lead to ridiculous increases in engine power by 1950.

(8) Streamlining is of the greatest importance to automobile builders; (*a*) because of the great reduction in air resistance possible and (*b*) because, above 30 m.p.h. it is the greater absorber of engine power.

(9) Rounding the corners of a sharp-edged body may reduce its air resistance to nearly one-half; but this reduction does not compare with that obtainable by streamlining front and rear, which may reduce it to one-tenth its original value.

(10) The conventional sedan has half as much resistance as the flat plate, said plate being one of the worst known examples of air resistance.

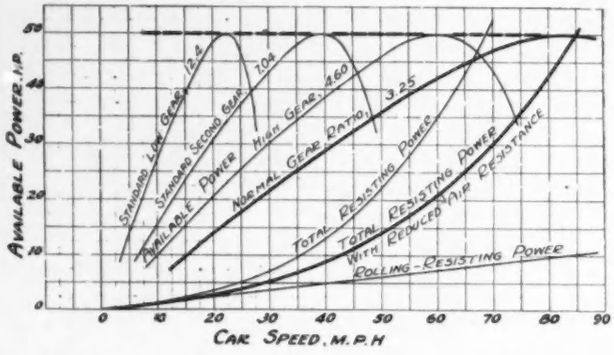


Fig. 36—Resistance and Power Available in Various Gears for a Small Car

(11) The conventional sedan due, first, to the shape of its body proper, second, due to the turbulence producing air-claws such as fenders, head-lamps, spare tires, bumpers, horns and the like, which are external to the body, has a much greater resistance than that of a rectangular box with properly rounded corners which will envelop the car in question.

(12) A five-passenger sedan may be constructed, streamlined front and rear, which will have only one-fifth the air resistance of a conventional sedan.

(13) While in the study of these models it was accomplished by means of a rear section of extreme length, both objectionable and entirely impracticable, other means of accomplishing the same results have been partially investigated.

(14) We believe that streamlined cars provided with variable-gear transmissions may be built to travel three times as far on a gallon of gasoline as the conventional sedan can do at normal touring speeds.

Suggestions Offered to Car Builders

The gist of this discussion may be summarized by making a few suggestions to the automobile builders, as follows:

(1) Remove all barnacles or wind-claws from the car. If they cannot be removed, build the body out to enclose them.

(2) Replace all sharp edges and corners with round edges and corners of generous radii.

(3) Build the front of the vehicle to bore a hole through the air with the least possible disturbance of the surrounding air.

(4) Build the rear of the body to lay the air back in place without eddies or turbulence.

(5) The shape of the ideal streamline form naturally provides space for housing the engine at the rear.

(6) The public is becoming streamline conscious and will welcome these changes at a more rapid rate than ever before. At this time it is particularly keen to accept changes which reduce operating costs.

(7) We must push our attack on the problem of the variable-speed automatic or semi-automatic transmission. The advantages offered by such a device are both numerous and important.

The difficulty in applying these suggestions is fully appreciated. It is certain that the public will spend far more money for its conception of beauty in a car even when it has less activity and a lower fuel mileage than it will for a streamlined monstrosity. But the beautiful car of a decade ago appears uncouth today. Moreover, the form of the streamlined car naturally offers greater opportunity to the artist who has the vision to appreciate it. A little engineering, a little art and some education of the public may accomplish much. The dove is streamlined by nature, so that it can fly with the least possible air resistance, and we call it beautiful. The day is coming when we shall drive streamlined cars and marvel at their beauty.

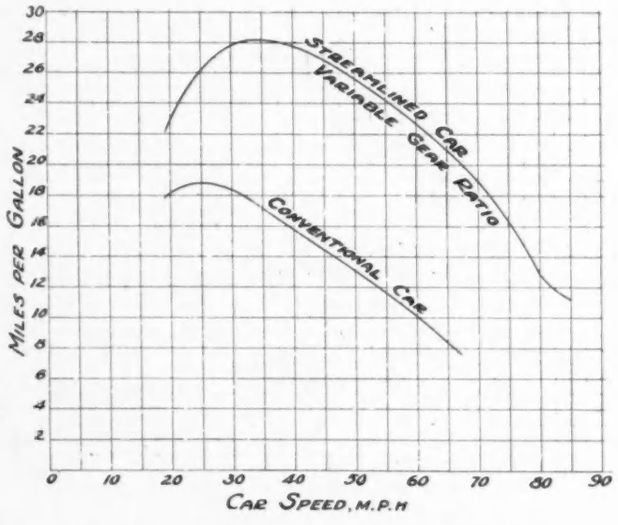


Fig. 37—Fuel Mileage with Variable Gear Ratio

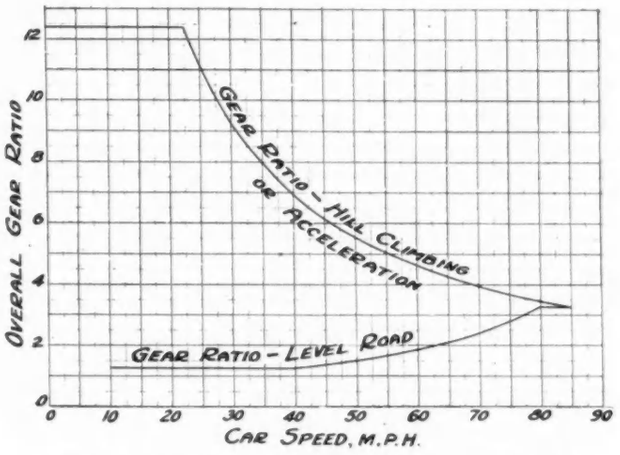


Fig. 38—Desirable Gear Ratios for Optimum Car Activity and Greatest Fuel Mileage

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SMALL tractors scarcely bigger than baby carriages powered with tiny motors can be built for economical operation on small farms of less than 20 acres, according to C. G. Krieger, engineer of the Detroit laboratories of the Ethyl Gasoline Corp., a member of the Society since 1930. Anti-knock gasolines and high compression engines permit the design of such units for high efficiency, he asserts.

What is virtually an untouched field of development lies open to the American farm machinery industry in the manufacture of tractors low in initial cost and upkeep, and small enough to be useful on farms where the big kerosene burning tractors of today find no place, declares Mr. Krieger, pointing out that more than 50 per cent of Americas six and one-half million farms contain less than 100 acres.

Additional Opinions Concerning Antiknock Research

SUPPLEMENTAL constructive comments by three eminent authorities on this subject include opinions regarding road-test correlation, in which the concept of comparison of cars on a "severity-level" basis is amplified; three statements of important considerations that should influence groups which may wish to test for themselves the correlation between the average performance of fuels in cars on the road and the ratings of these fuels in the laboratory by the C.F.R. Motor Method; and a somewhat different analysis of all the data secured at Uniontown, irrespective of fuels used on a particular car, to show that the authors' discussion need not be limited to reference-fuel data as set forth in the paper in the section devoted to "application to automotive-engine design."

Road-Test Correlation

D. B. Brooks[1]:—A study of Bureau of Standards road-test data obtained during the spring and summer of 1932 suggested that each make of automobile under normal road operation presents a characteristic severity of test conditions which affect the relative detonation tendencies of different fuels. Thus, some cars were found to give results comparable with those obtained in the laboratory by the C.F.R. Research Method, others were intermediate between that and the Motor Method, and some were lower than the last.

When the Uniontown data became available a similar study was made, using the Waukesha laboratory values. While different groups of cars were used in the two series of road tests, the results are in good agreement. Both sets of data show conclusively that different cars, operated under identical test procedure, subject fuels to test conditions which are more or less severe depending on design and operating factors characteristic of each car; both further indicate that the present Motor Method rates fuels within experimental error, in the order of their car ratings.

In analyzing the data, comparisons were made between: (*a*) car ratings and Motor-Method ratings; (*b*) car ratings and values deduced from the Research-Method and Motor-Method ratings by assuming that each car tends to "depreciate" all fuels by a characteristic *proportion* of the difference between the two laboratory ratings; and, (*c*) car ratings and values deduced by assuming each car to rate all fuels a constant amount higher or lower than does the Motor Method.

Before discussing the results, it is desirable to compare the above criteria and to note certain consequences of each. If all makes of automobiles presented test conditions on the road substantially equivalent in severity to those of the Motor Method in the laboratory, the differences obtained by each method of comparison should be small and roughly equal and would represent experimental error. If each car has a characteristic severity of test conditions, then the differences obtained by comparison (*b*) should be distinctly less than that by (*c*), and much less than that by (*a*). If the response to change in severity of test conditions is anomalous for the reference fuels *C*-6 and *A*-2, then differences by comparison (*c*) should be least. If comparison (*a*) gives the least residuals, or differences, all fuels will be rated in proper sequence by the Motor Method. If comparison (*c*) shows the least residuals, all fuels, except the two reference fuels and any fuels having a similarly anomalous response to change in severity of test conditions, will be rated in proper sequence by the Motor Method. However, if comparison (*b*) shows distinctly the least residuals, neither laboratory method alone can rate the fuels in proper sequence for all cars, as this will vary from one car to another.

The differences obtained by each method of comparison were treated statistically and the mean residual was found for each car and for each fuel as well as for the method. Residuals by comparison (*b*) were one-fourth, and by (*c*) one-third, of those by (*a*). The distinction is increased if the fuels are grouped according to the sensitivity of their ratings to a given change in severity of test conditions. The ratio of the mean residual for six fuels showing the greatest difference between Research-Method and Motor-Method ratings to the mean residual for five fuels showing the least difference was 2.33 by (*a*), 1.20 by (*b*), and 1.39 by (*c*). Thus, method (*b*) gives the lowest residual, although it is not so far below the value by (*c*) as to invalidate the latter altogether.

Another method of handling the results consists of correlating the mean residual for each fuel with the difference between its two laboratory ratings. The results are as follows:

Method of Comparison	(*a*)	(*b*)	(*c*)
Correlation Coefficient and Probable Error	$+ 0.908 \pm 0.173$	$+ 0.313 \pm 0.279$	$+ 0.504 \pm 0.262$
Random Probability	1.6×10^{-7}	0.28	0.08

The very high correlation method (*a*), of differences of car and Motor-Method ratings with corresponding differences of Research and Motor-Method ratings, together with the

[The paper on Antiknock Research—by C. B. Veal, H. W. Best, J. M. Campbell and W. M. Holaday—was printed in the March, 1933, S.A.E. JOURNAL, beginning on p. 105.]

[1] Automotive engineer, Bureau of Standards, City of Washington.

absence of correlation for method (*b*), confirms the hypothesis that each car has a characteristic severity of test conditions. This severity may differ widely for different makes.

The correlation for method (*c*) is doubtful. This indicates that, while there is a correlation, the magnitude of the effect is about equal to that of experimental error. This is quite fortunate as it follows that, while the Motor Method does not give ratings of the same value as do different cars, a car which is found to operate satisfactorily on any fuel of a given Motor-Method rating (except reference fuels and the like) will operate with equal freedom from detonation on any other fuel of equal rating, the foregoing being true within the experimental error of road rating by experienced operators.

The conclusions drawn are that:

(1) The severity of test conditions for the road rating of fuels under the Uniontown procedure has a characteristic level for each car.

(2) These levels vary, for 1932 cars, from that of the Research Method to an equal amount beyond that of the Motor Method.

(3) On the average, the Motor Method rates commercial fuels in the same sequence as do cars, within experimental error.

(4) As automobile-engine design is improved so as to utilize the antiknock value of available fuels to better advantage, conclusion (3) will become invalid unless the severity of the Motor-Method test-conditions is reduced.

(5) Two makes of cars give evidence that engine design can be improved to obtain conditions on the road no more severe than those of the Research Method.

Mr. Brooks Discusses Test Data

Several points raised since the Uniontown tests were made seem to warrant further comment. The following discussion is based on test data as published in the paper.

That the C.F.R. Motor Method will require periodic revision is apparent and was anticipated by the Committee. That it is not, or will not remain, perfect does not seem to constitute a valid argument against its adoption. If people waited for automobiles to be perfected before purchasing one, neither automobiles nor a detonation problem would be troubling us. More seriously, the A.S.T.M. distillation test was adopted long before anyone realized its inherent possibilities. There is already some evidence that the present detonation test methods, likewise, may have inherent possibilities for predicting car performance of fuels much beyond those realized at present.

Had each fuel been rated in every car in the Uniontown tests, valid conclusions readily could be drawn from the results. However, in one-third of the cases no rating was obtained, for one reason or another. In consequence, conclusions drawn from the results may be distorted unless allowance for this is properly made.

Average Value of "Depreciation" Ratios

Values of severity were obtained for each of 14 cars used at Uniontown by assigning the values 0 and 1 to the Research and Motor Methods, respectively, and then determining for each car the average value of the ratio of car "depreciation" to laboratory "depreciation."

These ranged from 0.27 to 2.33, the mean being 0.98. From these data the average severity of the road tests on a given Uniontown fuel can be found. Had each fuel been rated in every car, these average-severity values would all be 0.98. Actually, they range from 0.58 to 1.13. Fuels *RT*-9, *RT*-10, *UT*-6, *UT*-9, *UT*-10 and *UT*-11, were tested in cars averaging 0.78, while fuels *UT*-2, *UT*-3, *UT*-4, *UT*-5 and *UT*-7 were tested under average severity of 1.06.

Actually, when due allowance is made for the mean relative severity of tests, the Uniontown car-tests agreed within

Table 1—Comparisons Based on the Severity-Level Concept

Uniontown Car No.	Severity Level	Mean Laboratory Depreciation of Fuels Used in Car	Departure of Car Ratings from Research Rating (Octane Units)	Departure of Car Ratings from Motor Rating (Octane Units)	Maximum Deviations of Individual Ratings from Values by Motor	Maximum Deviations of Individual Ratings from Values by Severity Level[a]
1	0.72	3.9	—2.8	+1.1	6.7	3.9
2	2.33	3.7	—8.7	—5.0	12.3	3.6
3	0.97	2.8	—2.7	+0.1	2.8	3.1
4	0.27	4.3	—1.2	+3.1	8.1	2.0
5	1.63	2.4	—3.9	—1.5	2.8	2.5
5′	2.20	3.6	—8.0	—4.4	6.7	2.2
7	1.00	3.9	—3.8	+0.0	5.0	5.0
8	1.08	3.7	—4.0	—0.3	5.0	4.2
10	0.71	2.6	—1.9	+0.7	2.5	3.4
10′	0.43	4.3	—1.9	+2.4	4.8	3.1
12	0.58	2.7	—1.6	+1.1	2.8	0.8
13	0.61	5.7	—3.5	+2.2	4.5	4.2
14	0.42	4.2	—1.7	+2.4	7.3	3.1
15	0.72	5.9	—4.3	+1.6	4.2	3.9
				Maxima =	12.3	5.0
				Average Deviations =	2.54	1.38

[a]Obtained as follows: Predicted Car Value equals the Research Value minus the Car-Severity Factor times the difference between the Research and the Motor Values.

0.5 octane unit with the laboratory tests, except in two cases: *RT*-10 was rated 0.8 octane unit low, and *UT*-11 was rated 1.1 octane unit high, by the cars.

Table 1 illustrates the variation in severity level from car to car, the variation in mean "depreciation" of the fuels used in the several cars, and the improvement in precision of predicted car values made possible by use of the severity-level concept.

Road Detonation Tests

H. K. Cummings[2]:—It is to be expected that many groups will wish to test for themselves the correlation between the average performance of fuels in cars on the road and the ratings of these fuels in the laboratory by the C. F. R. Motor Method. In this connection three considerations are believed to be important:

(1) In the interests of uniformity, the road evaluation of knock always should be made according to the procedure developed at Uniontown, wherein the fuel under test and the two reference blends are compared at the respective car speeds giving maximum knock.

(2) When practicable, fuels used in road tests should be rated in the laboratory by the Research Method as well as by the Motor Method to note the effect on the rating of a definite change in the severity of test conditions. If equipment for Research-Method ratings is not available, a definite but smaller change in the severity of test conditions can be had by rating the fuels at 900 r.p.m. with and without mixture heating.

(3) In view of the considerable experimental error in the road evaluation of knock for a given fuel in a given car and the variation of average ratings from car to car, one must guard against drawing positive conclusions from too few data.

Fuel-Sample Analysis Available

With reference to consideration (1), the Uniontown Hill is not essential to the procedure there adopted. The condition of slow acceleration over a suitable speed range can be reproduced exactly on a level road by means of a variable drag such as a traction dynamometer, or on a chassis dynamometer of sufficient capacity. The Bureau of Standards recently has assembled, at very slight cost, an electric traction dynamometer well adapted to such work. An obsolete street-car motor was mounted on the stripped chassis of a second-hand truck in place of the gasoline engine. The series-wound field of the motor was detached from the armature circuit and connected to a 6-volt storage battery through a slide-wire rheostat, to give load control. The armature output is passed through cast-iron grids in series which have a total resistance of about 3 ohms. This traction dynamometer is coupled to the test car by a drawbar which also steers the dynamometer.

Regarding considerations (2) and (3), we wish to suggest that all groups undertaking comparative road tests send the road ratings together with both laboratory ratings (or a 1-gal. sample of each fuel used) to the Bureau of Standards for analysis. If this is done, statistical data based on a greater variety of cars and fuels than are likely to be tested by any one organization can be made generally available at an early date. Information as to car models and fuel brands will be

[2] M.S.A.E.—Chief, automotive powerplants section, Bureau of Standards, City of Washington.
[3] M.S.A.E.—Standard Oil Development Co., New York City.
[4] See S.A.E. JOURNAL, March, 1933, p. 114.

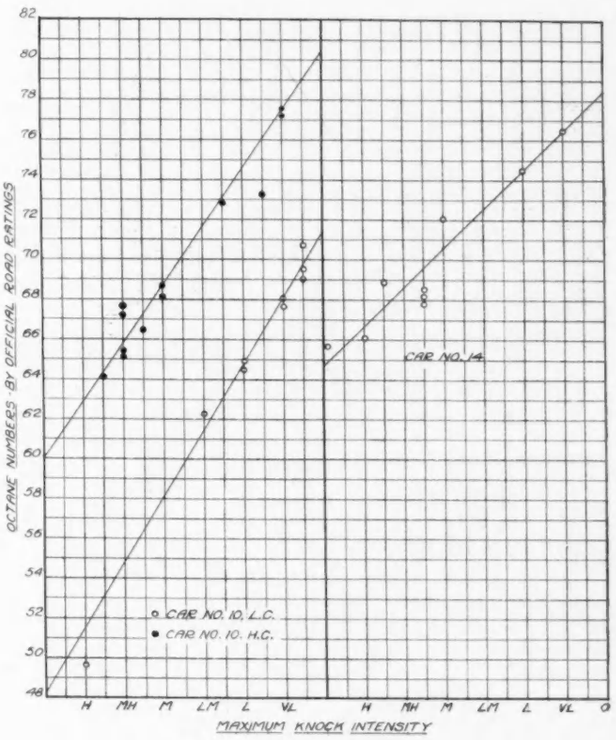

Fig. 1—Change in Maximum Knock Intensity with Road Octane Numbers

useful in classifying the data; but, of course, particular cars and fuels will not be identified in any report released for publication.

The data obtained in the summer of 1932 indicate, as Mr. Brooks' analysis has pointed out, that the commercial fuels were rated in substantially the same order, within the experimental error of the road ratings, by all the test cars used at Uniontown. The object of the proposed statistical study is to test this and other tentative conclusions with respect to a wide variety of cars and fuels.

Another Analysis of Uniontown Data

A. E. Becker[3]:—My remarks are directed primarily to the section Application to Automotive Engine Design of the paper. The authors have limited their discussion to reference-fuel data and have prepared a tabulation[4] showing the octane-number requirements of the various cars for knock-free operation when using these reference fuels.

A somewhat different analysis of all the data secured at Uniontown, irrespective of fuels used on a particular car, shows that no such limitation need be made. Referring to Fig. 5 of the paper it will be noted that fuel *RT*-11 was found to produce a medium knock at 40 m.p.h., the speed of maximum knock for this fuel: From similar curves the maximum knock ratings determined during the tests and the corresponding maximum knock intensities were assembled for each car, regardless of when or by whom the fuels were tested. Where a fuel was tested in a car by more than one crew, average values were used.

These data are shown for two of the cars used, in the accompanying Fig. 1, where the coordinates are maximum knock intensity and road ratings expressed as octane numbers

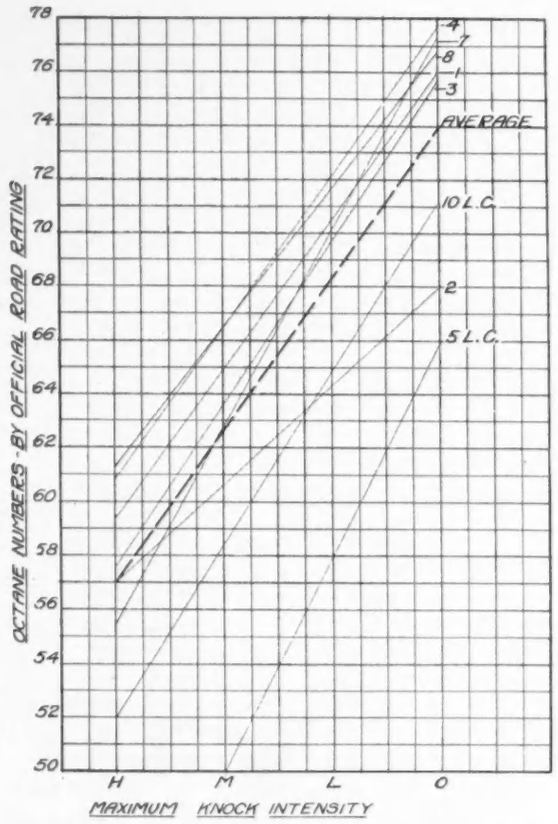

Fig. 2—Change in Maximum Knock Intensity with Road Octane Numbers for Low-Compression Cars

(Motor Method). In each of the three cases shown the data are quite well represented by straight lines. Similar results were obtained for the other cars used, as indicated in Figs. 2 and 3, herewith, for the low and the high-compression cars respectively.

Several points of significance, particularly to the engine designer, are obvious:

(1) The tendency for a car to produce objectionable knock intensity in its critical speed range increases as the octane number by the C. F. R. Motor Method decreases, regardless of the types of fuels used.

(2) The octane-number requirements of the test cars are readily determined. It is only a question of deciding upon the degree of knock that a customer would regard as being objectionable. For car 10, Fig. 1 shows that a fuel of octane No. 65 will give a light knock with the low-compression head but that one of octane No. 71 is required for absolute elimination of knocking tendency throughout the speed range. Reference to Figs. 2 and 3 will show that octane-number requirements for complete elimination of knock agree with the values given in the tabulation in the paper.

(3) The sensitivity to change in octane number is surprisingly equivalent for most of the cars, notable exceptions being Cars 2 and 14. In these a given increase in the octane number of the fuel is so much more effective in decreasing knocking as to suggest that a study of the reasons would prove most instructive. If due to richness of mixture or to a retarded spark, it is likely that it has been accomplished at an undesirable sacrifice of power and economy. If due to design factors, such as valve timing or a more uniform air-fuel ratio throughout the speed range, it would appear to be a highly desirable accomplishment.

(4) The fuels supplied by the petroleum industry must of necessity represent a compromise. Fig. 2 shows that for the low-compression cars a fuel of octane No. 66 will give knock-free operation in Car 5 and a medium knock in Cars 4 and 8, with intermediate tendencies for the other cars.

(5) Raising the compression of an engine, Fig. 1, without altering other factors such as the temperature conditions under the hood, results in a shift of the octane-number knock-intensity curve in the direction of higher octane-number requirements. Undoubtedly this is due to increased temperatures at the end of the compression stroke. It shows the need of keeping temperatures under the hood at a low value, which is in line with one of the main requirements for avoiding vapor lock.

Fig. 1 shows also that the data for both the low and the high-compression heads of Car 10 are equally well represented by straight-line curves. These facts do not indicate that better engine design will result merely in a fuller use of the antiknock qualities which some fuels have at lower combustion-chamber temperatures, as W. M. Holaday has suggested, but that the designer will immediately make full use of the advantage gained by raising the compression ratio of his engine a corresponding amount.

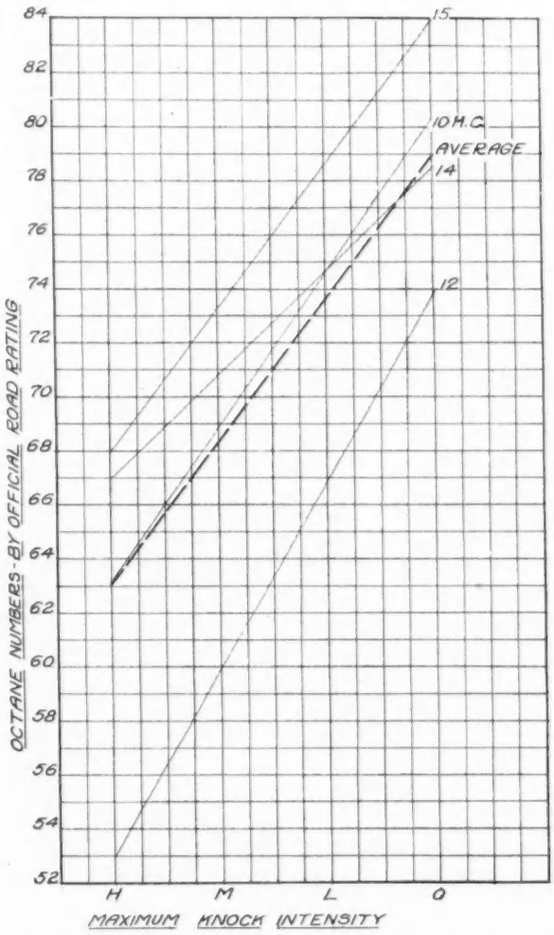

Fig. 3—Change in Maximum Knock Intensity with Road Octane Numbers for High-Compression Cars

An Analysis of Tires and Wheels as Causes of "Tramp"

By O. E. Kurt

THIS paper presents the solution of the factors in tires and wheels that cause a particular type of front-end vibration termed "tramp," which is a vertical vibration of the front axle accompanied by a small degree of simultaneous oscillation of the wheel assembly about the king pin. This vibration in turn sets up the disturbance of the body and chassis. The front-axle vibration is caused by the unbalance and variation in rolling radius of the rotating front-wheel assemblies. The theoretical action of these two factors is developed in detail and supported by experimental results.

The foregoing two factors act independently. The resultant of the two periodic forces which they set up depends on their phase relationship. The forces due to unbalance are in the same direction as the forces due to variation in rolling radius and enhance their effect when the center of gravity of the rotating assembly is in the region of minimum rolling radius; they are in the opposite direction and nullify the forces due to variation in rolling radius when the center of gravity is in the region of maximum rolling radius.

The amount of unbalance and variation in rolling radius and the amplitude of the resulting axle vibration necessary to cause tramp varies with different automobiles. They can be reduced by increasing the uniformity of tire-wheel assemblies and their effectiveness can be reduced by proper body design. The problems of the tire, wheel and automotive engineers in eliminating this vibration are discussed.

THE problem of front-end vibration was directed to the attention of the tire engineer when the first balloon tires were found to increase the susceptibility of automobiles to this disturbance. Their contributions which have appeared in the publications of the Society manifest the interest they have taken in this problem, which interest has continued to the present time. It is hardly necessary to point out to anyone acquainted with this branch of tire engineering that this problem still requires consideration in the proper design of automobiles and is one of those the solution of which is necessary for the satisfactory adoption of the even lower inflation pressures of super-pneumatic tires.

The scope of this paper, although it must consider the general problem of front-end vibration, is restricted to that particular type of vibration which is commonly designated as "tramp." To analyze this phenomenon and discover its causes and effects, its nature must be defined. Tramp has been variously described as a vibratory movement of the front axle, fenders, lamps and radiator, depending on which of these was observed. Recently it has been recognized that the body vibrations are only the result of the axle vibrations and that their interrelationships depend on the design of the automobile. It is evident, therefore, that the body vibrations which are the objectionable feature of tramp could be eliminated either by stopping the axle vibrations or by preventing the transference of the axle vibrations to the body. The mechanism of the transfer of the axle vibrations to the body is essentially a problem in automobile design and is being solved by the automotive engineer. The manner in which the axle vibration is caused, and especially the role of the tires and wheels in its excitation, is a problem for the tire engineer and is the subject of this paper.

On the basis of observations, "axle tramp" is defined as essentially a vertical vibration with the small degree of "wheel wabble" or "wheel shimmy" which results from the simultaneous oscillation of the spring and drag link about two different centers. Because of this small amount of shimmy, tramp has also been termed "high-speed shimmy." On the other hand, "shimmy" consists mainly of the oscillation of

[This paper was presented at a meeting of the Indiana Section of the Society. Mr. Kurt is tire-development engineer for the U. S. Tire Co., Inc., Detroit.]

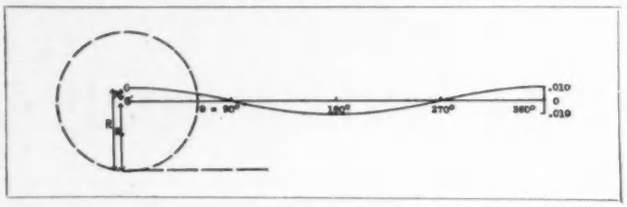

Fig. 1—Effect of an Off-Center Axis

The axis or center of rotation of the assembly is located at C and the true center of the assembly at C'. The displacement (magnified) of the axis from the true center is CC', and is designated A. The rolling radius is denoted by R, and $R = R_0 + A \cos \theta$.

the wheel assembly about the king pin with the small amount of vertical hub movement which it causes due to the caster and camber of the king pin. Tramp and shimmy may occur simultaneously, but they differ enough in characteristics to be distinguishable. A certain amount of each always accompanies the other, depending on the mechanical design of the front end.

Experimental evidence is presented in this paper to show that the factors in the rotating tire-wheel assembly which are responsible for tramp are its runout and unbalance. The theoretical action of these two factors is developed in detail and is supported by the experimental data. The methods of investigation which were employed did not postulate special characteristics to the test cars employed. Therefore the observations, the theory based on these observations and the conclusions which are drawn, should apply at least qualitatively and generally to any make of automobile which employs the present regular front-end suspension and steering mechanism. The only specific factors which differ for each automobile are the characteristic frequency and the amplitude of axle vibration necessary to cause observable body vibrations.

Characteristics of Tramp

Certain characteristics were found to distinguish the vibratory movements which occur in tramp from those of other front-end disturbances. Tramp is usually noticed first in vibration of the body, headlights, radiator or fenders, depending on the construction of the car. In some cases these may be sufficiently well insulated so that they will not be noticed to vibrate until the axle vibration has reached sufficient amplitude to be felt in the steering wheel. In the investigation of this problem the front axle was observed and the following characteristics were noted:

(1) Only a vertical component could be detected in the axle movement. The front wheels oscillated about the king pin, but only by the slight amount which could be explained as due to the spring and drag-link arc relationships.

(2) The frequency of the tramp vibration corresponded to the frequency of rotation of the front wheels, thereby indicating that the vibration is caused by periodic forces set up by the rotating assembly.

(3) The body vibrations occurred at a definite speed within a range of about 3 m.p.h. At higher and lower speeds they disappeared completely. This indicated that the body was vibrating in resonance with the axle.

(4) The body vibrations were most regular and noticeable on smooth pavement, probably because external irregular vibrations from the road interfere with the resonance effect.

From these observations it was evident that the axle vibration was caused by periodic forces having vertical components of a definite frequency. Of the following forces which can be conceived as being set up by the rotation of the tire-wheel assembly itself and by its reaction with the road surface only, the first two meet these requirements; that is, forces due to

(1) Variation in rolling radius; periodic

(2) The unbalance of the rotating assembly; periodic

(3) Irregularities in the road surface; spasmodic

(4) Gyroscopic action of the disturbed rotating system; periodic but of varying frequency

Necessarily, such disturbances as might arise due to worn parts, irregularly worn tires and loose parts, were eliminated by running all of the tests with new cars on new tires. The observations and the conclusions up to this point leave the variation in rolling radius and the unbalance of the rotating assembly as most important and to be investigated first. The effect of each of these will now be analyzed separately.

Variation in Rolling Radius.—The rotating assembly consists of the hub and brake drum, the wheel and the tire. The rolling radius is the distance from the center of the axle to the road surface. It may vary in length as the assembly rotates, due to any of the following causes:

(1) An off-center position of the hub would cause a simple variation in rolling radius; that is, $R = R_0 + A \cos \theta$ as shown in Fig. 1.

(2) There is always some deviation from a true circle in the rim bead-seat surface. This is called the radial runout and can be measured by placing a runout indicator on this surface as the wheel is rotated on the axle. A typical result of such a measurement is shown in Fig. 2.

(3) Small variations in the deflected cross-section of a tire can be similarly measured by placing an indicator against an extension on the axle and deflecting the tire in various rotational positions mounted on a perfectly true wheel-and-hub assembly. A typical rolling-radius variation as measured with a tire on a rim having a small amount of runout is shown in Fig. 3.

The forces caused by a variation in rolling radius can now be considered. The total weight supported by the axle causes downward acceleration of the axle. This is balanced by the upward force F_1 exerted by the road surface on the tire at the deflected area as shown in Fig. 4. When these two forces are balanced to equal each other, we have

$$F = F_1 + F_2 = 0 \qquad (1)$$

and there is no vertical acceleration of the axle.

As the assembly rotates, the equilibrium rolling radius varies; but, due to inertia, there is no immediate change in the actual rolling radius. The effect is the same as a corresponding variation in deflection and causes a variation in the upward force F_1. The forces which are involved can be calculated if the relation between F_1 and the rolling radius R is known. In Fig. 4, the rolling radius R is identical with CD. An increase in the deflection h produces a decrease in R. Hence we have

$$dR = -dh \qquad (2)$$

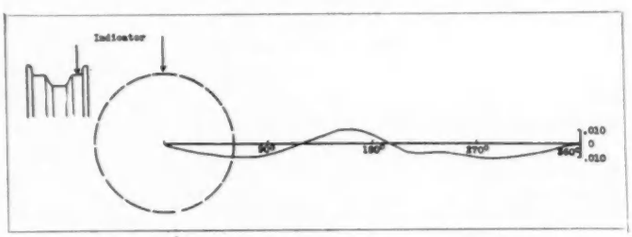

Fig. 2—Typical Low Radial Runout on the Bead Seat of a Wheel

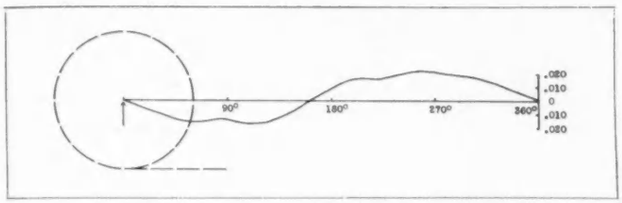

Fig. 3—Typical Variation in the Rolling Radius of a Tire-Wheel Assembly

The relation between h and F_1 is given by the load-deflection curve, the slope of which is constant over the small changes in deflection involved. In other words,

$$F_1 = K_1 h + K_2 \quad (3)$$

$$(dF_1/dR) = (-dF_1/dh) = -K_1 \quad (4)$$

For example, as shown in Fig. 5, a 6.00-18 four-ply U. S. Royal tire on the front of a test car is deflected 0.96 in. by the standard load of 1100 lb. and, from the slope of the experimental load-deflection curve, the deflection changes with change in load at the rate of 1 in. per 1550 lb. Then,

$$(dF_1/dR) = (-dF_1/dh) = -1550$$
$$dF = -1550\, dR \quad (4a)$$

As the assembly rotates and the regions of greater and less rolling radius ("high" and "low" parts) make contact with the road, there are alternate increases and decreases in F_1. The reaction consisting of upward and downward acceleration of the axle is not immediate because of the inertia of the system, but has the same period as the rotation of the assembly.

Effect of Unbalance.—If the assembly is not balanced its rotation causes an unbalanced centrifugal force which, acting in the direction of the center of gravity of the rotating assembly, rotates with the assembly and alternately increases and decreases F_2 and consequently the resultant F. For example, as shown in Fig. 6, to calculate the centrifugal force caused by an unbalance of B in-oz. in the rotating assembly under the foregoing consideration at a speed of 62 m.p.h., we have:

$$CF = 4\pi^2 (mr/T^2 g) = 0.898\, B \quad (5)$$

where

T = period of rotation $= 2\pi R/v$

$mr = B$ in-oz. $= B/(12 \times 16) = 0.0521\, B$ ft-lb.
$=$ unbalance of the mass, m, acting at a distance r (6)

$v = 62$ m.p.h. $= 62 \times 1.467 = 90.95$ ft. per sec. (7)

$R = 14.65$ in. $= 1.221$ ft. $=$ rolling radius (8)

The change in F which is produced by the centrifugal force, due to an unbalance of 1 in-oz. acting in first one direction and then in the other, is $dF = dF_2 = 2 \times 0.898 = 1.80$ lb.; and, for B in-oz. of unbalance, the change in F is $dF_1 = 1.80\, B$ lb.

According to Equation (4*a*), this is equivalent to the force caused by an instantaneous change in rolling radius of

$$R = F/-1550 = -1.80\, B/1550 = -0.00116\, B \text{ in.} \quad (9)$$

The force F_1 varies, due to variation in rolling radius, in an irregular manner, as the assembly rotates. Measurements made of rolling radius indicate that it usually reaches only one minimum and one maximum per rotation. If the variation in rolling radius were due alone to the hub being off center, as shown in Fig. 1, then the variation in F_1 could be expressed by a simple equation; but only if the hub can be assumed to remain at rest for

$$h = S - R \text{ and} \quad (10)$$
$$h_0 = S_0 - R_0 \quad (11)$$

where S equals the standing height or radius of undeflected tire, and S_0 and h_0 are the values for the standing height and deflection corresponding to R_0. In the case under consideration,

$$S = h_0 + R = h_0 + R_0 + A \cos \theta$$
$$S = S_0 + A \cos \theta$$

If the rolling radius were kept constant at R_0 by not allowing the hub to move in a vertical direction, then the deflection would change thus:

$$h = S - R \quad (10)$$
$$h = S_0 + A \cos \theta - R_0 = h_0 + A \cos \theta$$

Since $F_1 = K_1 h + K_2$ according to Equation (3), we have:

$$F_1 = K_1 (h_0 + A \cos \theta) + K_2 = K_1 h_0 + K_2 + K_1 A \cos \theta$$
$$= F_0 + K_1 A \cos \theta \quad (12)$$

In other words, the upward force would vary simply and periodically. Under actual operating conditions the hub would move and the problem is a complicated one in which the sprung and unsprung weight must be considered as well as the elastic properties of the springs, tires and moving body-parts.

Since the variation in rolling radius is always irregular, it would be quite impossible to make a rigid calculation for any particular case. For the purpose of this problem it is sufficient to know that the accelerating force caused by the variation in rolling radius is never as large as calculated with the assumption of a stationary hub. The force will vary regularly with each rotation and might be expressed as

$$F_1 = K_3 + f(\theta_1) \quad (13)$$

The force F_2, on the other hand, varies in a very regular manner. If θ_2 is the position angle between the center of gravity and the direction of F_2 from the center of rotation of the assembly, then the effective component of the centrifugal along F_2 is $CF \cos \theta_2$ or

$$F_2 = -K_3 - CF \cos \theta_2 \quad (14)$$

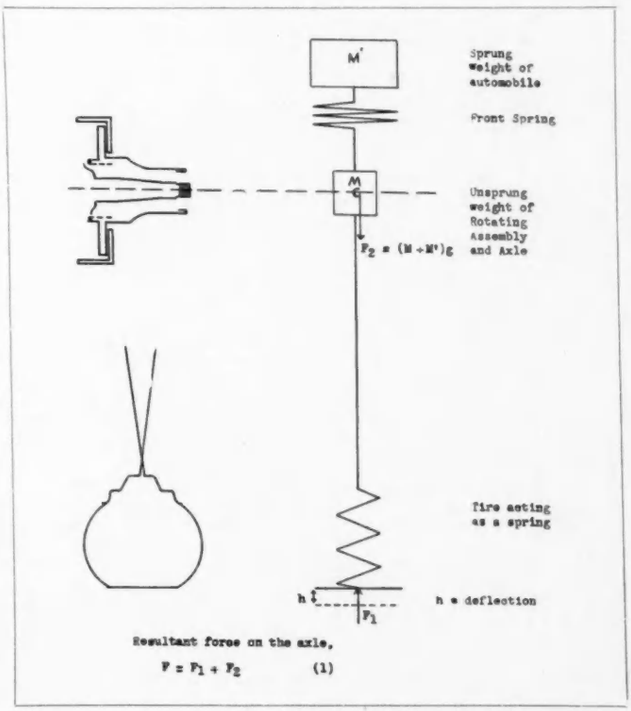

Fig. 4—Vertical Forces Involved in Axle Vibration

The variation in the resultant force acting on the hub is shown by the Equation

$$F = F_1 + F_2$$
$$= K_2 + f(\Theta_1) - K_3 - CF \cos \Theta_2$$
$$= f(\Theta_1) - CF \cos \Theta_2 \qquad (15)$$

The resultant force depends, therefore, on the amount of variation in rolling radius and unbalance and, in addition, on the difference between Θ_2 and Θ_1. For Θ_2 and Θ_1 change at the same constant rate; so there is a constant difference in phase between the periodic effects of the variation in rolling radius and the unbalance. The difference in Θ_1 and Θ_2 or phase relationship between F_1 and F_2 is just as important as either the variation in rolling radius or unbalance. F_1 and F_2 have the same period; namely, that of the rotation of the assembly. On the phase relationships depends whether they shall increase alternately or together. *The forces causing acceleration of the axle in a vertical direction, due to variation in rolling radius and unbalance, are independent and may enhance or nullify each other.*

More simply stated, if the unbalanced weight of the tire-wheel assembly is placed at the position of maximum rolling radius, the effects of these two factors will neutralize each other; if the unbalanced weight is placed at the position of minimum rolling radius, the effects will enhance each other. This is illustrated in Fig. 7.

References can be found in the reports of previous investigators to runout and unbalance as causes of tramp and shimmy, and it is common knowledge that these are the two most evident factors in the rotating assembly causing tramp. However, an investigation of the limiting values of runout and unbalance which must be exceeded to cause tramp, without regard to the relative positions of the two factors in the rotating assembly, leads to anomalous results. In fact it was the disparity in the results of previous tests in which the relative position of these two factors was disregarded that caused this investigation to be initiated.

Experimental Results

All of the final tests were run on a 1932 sedan which had a high susceptibility to tramp. Two other models of sedans used in preliminary tests had a much lower susceptibility to tramp, but the data are not sufficiently exact to present a comparison. The variation in rolling radius was measured on a number of tire-wheel assemblies previously selected to afford a range of values.

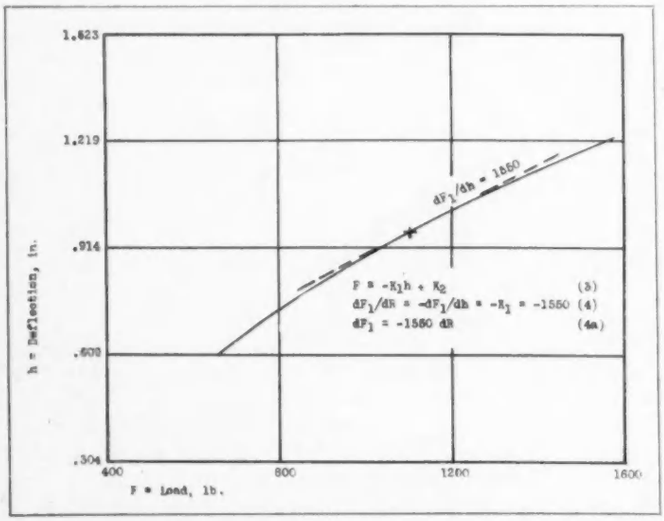

Fig. 5—Load-Deflection Curve for a 6.00-18 Four-Ply Tire at an Inflation Pressure of 36 Lb. per Sq. In.

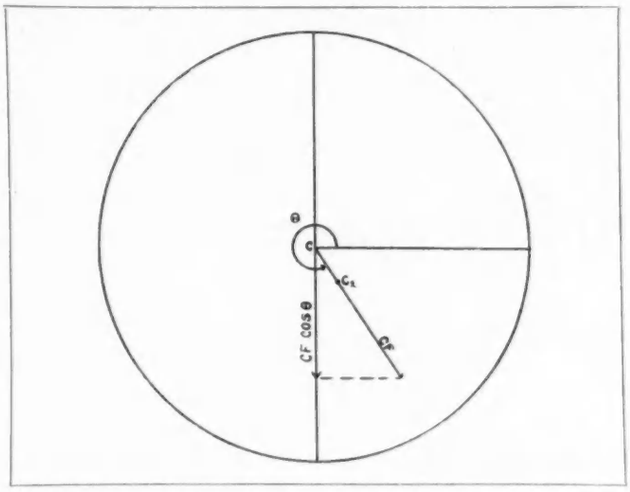

Fig. 6—Vertical Component of Centrifugal Force Due to Unbalance

The center of rotation is at C; the center of gravity, at C_1. The centrifugal force due to unbalance is represented by CF, and $CF \cos \Theta$ is the vertical component of centrifugal force.

One of these having a low variation in rolling radius was placed on the left front hub, and statically balanced; the remainder were run consecutively on the front hub. Balance lugs were placed on each wheel at the positions of maximum and minimum rolling radius and adjusted to just give a certain small amount of tramp. Results on the last six runs are given in Fig. 8. In accordance with the theory, a larger amount of unbalance is required to cause tramp when placed at the position of the maximum rolling radius than when placed in the position of minimum rolling radius. When placed in the position of maximum rolling radius the unbalance required increases with the variation in rolling radius, but when placed in the position of minimum rolling radius the unbalance limit decreases as the variation in rolling radius increases.

In these results with one front-wheel assembly balanced and having a variation in rolling radius of only 0.021 in., the limits for the other assembly were 0.056-in. variation in rolling radius when balanced and 0.018-in. variation in rolling radius when unbalanced 8 in-oz. The equations of the two limiting lines in Fig. 8 follow. The unbalance of the assembly in the position of minimum rolling radius enhanced the effect of the variation in rolling radius. The unbalance required to cause tramp increased as R, the variation in rolling radius, decreased thus:

$$0.00467\,B + \Delta R_{min} = 0.056 \qquad (16)$$

With the off-balance in the position of maximum rolling radius, it opposed the effect of the variation in rolling radius and the unbalance required to cause tramp increased with increased variation in rolling radius thus:

$$0.00175\,B - \Delta R_{max} = 0.021 \qquad (17)$$

Theoretically, according to Equation (9), Equations (16) and (17) should be

$$0.00116\,B + \Delta R_{min} = 0.060 \qquad (18)$$
$$0.00116\,B - \Delta R_{max} = 0.060 \qquad (19)$$

Thus the effect of the variation in rolling radius is not as great in the experimental equation as it is in the theoretical equation, and this agrees with the prediction made in deriving Equations (12) and (13) for this effect. The accuracy of the data in Fig. 8 is limited by the fact that the end point in the tests, namely, the presence of a certain small degree of tramp, was not sensitive to less than 5 in-oz. under the most favorable conditions. In running the tests for tramp, it was necessary to run all tests over the same identical road surface. The curves are represented as straight lines within the limits of error.

For these tests the variation in rolling radius was measured directly. The *radial runout* of the tire-wheel assembly is a more convenient measure of the same factor. It is obtained by placing an indicator against the crown of the inflated tire and rotating the assembly. If the radial runout curve is smoothed to exclude all irregularities which occupy less than one-tenth of the circle, the resulting radial runout is very nearly equal to the variation in rolling radius. A comparison of rolling radius and radial-runout curves on a typical tire-wheel assembly is presented in Fig. 9. Radial-runout measurements were employed in practically all of the tests and in the study of the causes of variation in rolling radius.

Rolling-Radius Variation and Unbalance

Having established the causes of an undesirable phenomenon, there remain the problems of either reducing these causes or rendering them ineffective. The former requires the attention of the wheel and tire engineers; the latter, the attention of the automotive engineers. With cooperation and a reasonable contribution by each, the problem is solved. Reasonable limits must be specified, based on the costs involved. This is the history of any problem.

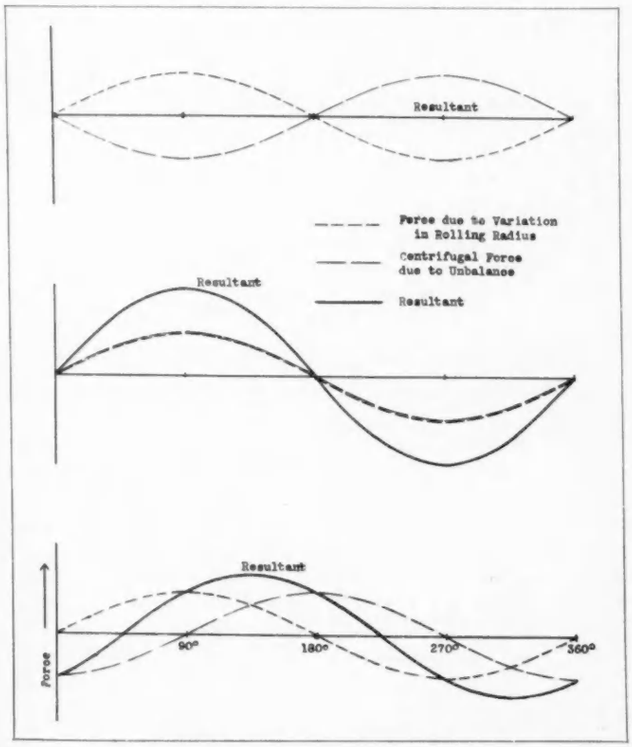

Fig. 7—Composition of Forces Due to Variation in Rolling Radius and Unbalance

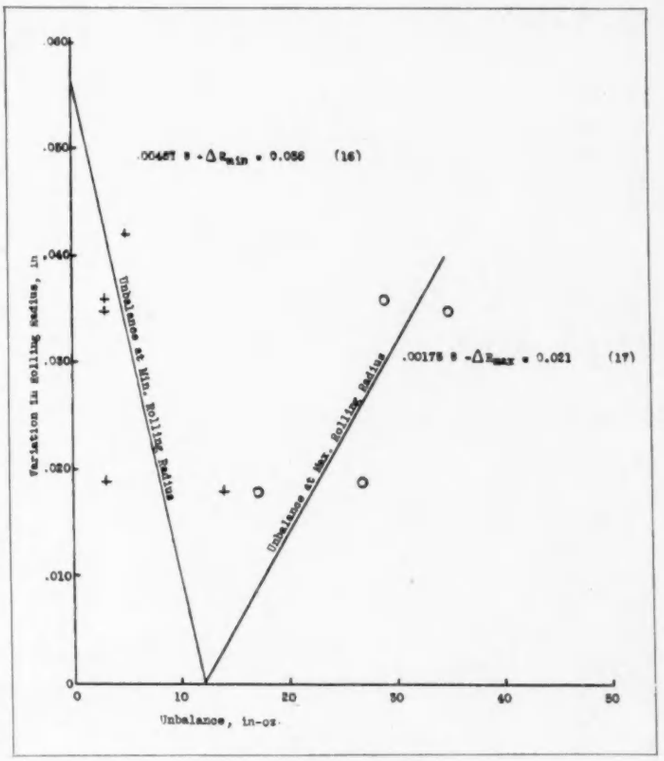

Fig. 8—Results of the Last Six Tests

The relative contributions of the tires and wheels to the causes of tramp can be analyzed simply. As regards the unbalance of the tire-wheel assembly, it is simply the vector sum of the unbalance of each of the parts—the hub, brake drum, wheel and tire—and, therefore, depends on the value of each of these and their relative positions. A reduction of the unbalance of each part or their arrangement to neutralize each other is necessary. Only the static balance is considered. The degree of dynamic unbalance necessary for a statically balanced assembly to cause tramp is many times greater than that which is regularly present.

The variation in rolling radius can be similarly ascribed to the deviations of each part and the manner in which they are assembled. To determine the effect of the tire alone, it should be measured on a true rim having no runout. The smooth curve showing the effective runout for a tire on a regular wheel is approximately the summation of the curves for the wheel bead-seats and the tire on a true wheel. The effective maximum runout of the assembly is usually less than the sum of the wheel and tire separately, depending on their relative positions.

The radial runout of the assembly is largely due to the off-center position of the hub. The irregularities and eccentricity of the rim are largely overcome by the tendency of the tire to form a true circle when inflated. The radial and lateral runout of the rim both produce an effect in the radial runout of the assembly, but the effect of the latter is small. Some have considered the lateral runout of the tire, wheel and assembly, as important as the radial runout, but its only disturbing effect should be to cause dynamic unbalance. The effect of lateral runout of the assembly in these tests was too small to be observed.

The runout caused by the tire can be ascribed to lack of

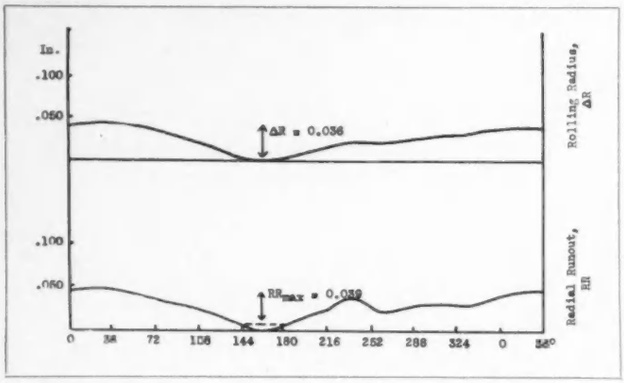

Fig. 9—Comparison of Rolling Radius and Radial Runout of a Tire-Wheel Assembly

uniformity. In fact, uniformity is the essential quality necessary in every part of the assembly to reduce both unbalance and runout. The cost of reducing the causes of tramp in tires and wheels is the cost of reducing tolerances and increasing uniformity. These can be accomplished only by improved methods of manufacturing. Uniformity has been accomplished in tire manufacturing by the adoption of mass-production methods. The flat-band method of tire building may be mentioned as one of the outstanding developments aiding the adoption of such methods.

Increased uniformity is the contribution of the tire and wheel suppliers. The contribution of the automotive engineer in turn is improvement in design to reduce the effectiveness of axle vibration. Each automobile has a set of limits. If the unbalance and rolling-radius variation exceed these limits, the automobile will tramp. These limits vary widely from one model to another.

As an example of this variation, one of the 1932-model sedans used for tests did not tramp with a balanced assembly unless the radial runout exceeded 0.080 in., or with an assembly whose runout was 0.020 in. unless the unbalance exceeded 40 in-oz., as compared with another whose limits were 0.060 in. for a balanced assembly and 14 in-oz. for a radial runout of 0.020 in. These limits are for one front assembly with the other balanced and having a radial runout of about 0.040 in. for the former automobile and 0.025 in. for the latter. Obviously, the latter would be more apt to tramp than the former on regular equipment, and the engineer equipping the latter would set up much stricter specifications on the tire-and-wheel equipment. Only cooperation and a mutual understanding of the problem involved will enable the solution of such a situation.

Extreme Pressure Lubricants

AT the present, the load-carrying properties of extreme pressure lubricants can be summed up as follows:

Most of the materials ordinarily known as composing extreme pressure lubricants for steel on steel use may be classified in the following group:

(1) Sulphur plus mineral oil, with or without the addition of saponifiable oil.

(2) Saponifiable oil treated with sulphur chloride and mixed with mineral oil.

(3) Mineral oil containing lead soap with or without the addition of added sulphur.

Extreme pressure lubricants are simply ordinary lubricants to which some suitable chemical materials have been added. Under ordinary conditions, extreme pressure lubricants act as ordinary mineral oil, but, under more severe conditions of load and speed, where mineral oil fails to lubricate, the chemical material in the lubricant acts on the steel to form an entirely different kind of a lubricating film that will resist scoring. The mineral oil, under these conditions, acts simply as a carrier for this chemical and as a cooling agent.

We would like to call particular attention to two points:

(1) In this discussion of extreme pressure lubricants we have been considering the lubrication of steel on steel, and the lubricants desirable for steel on steel may or may not be desirable for steel on bronze such as we have in worm gears.

(2) There is great danger of any rear axle lubricant thickening in service if operated at very high temperatures. The problem of extreme pressure lubricants should not be confused with that of thickening of the lubricant, and one should not attempt to cure the trouble of thickening in service by simply substituting an extreme pressure lubricant.

The commercial situation in regard to extreme pressure lubricants may be summarized as follows:

Extreme pressure lubricants are at the cross-roads. Many of the refiners are assuming the attitude that extreme pressure lubricants are not needed at the present time, and if and when they are required, they will produce them, while the automotive manufacturers are hesitating to introduce gear designs which require extreme pressure lubricants for satisfactory performance in service until these lubricants are universally distributed and are available at all filling stations.

Before the new types of rear-axle gears, which operate under more critical conditions of tooth loading and rubbing velocity, can become commercial for cars in large production, extreme pressure lubricants must be standardized in terms of load-carrying ability and made available for distribution.

A Paradoxical Situation

Our laboratories are in an embarrassing position. We are urging the petroleum industry to develop better extreme pressure lubricants and to standardize on methods for testing, and at the same time we are doing everything within our power to design and develop gears which do not require the use of extreme pressure lubricants for satisfactory operation in service. We have available gear designs which require the use of extreme pressure lubricants, but we shall continue to concentrate on designs which will operate on straight petroleum lubricants until the commercial situation on extreme pressure lubricants makes them available to the car owner.

Gear development is stalemated. Progress can only be made through cooperation by the petroleum industry in developing and supplying the necessary lubricants of high load-carrying capacity—Excerpts from conclusions of a paper presented at the Pittsburgh Section Meeting, March, 1933 by H. C. Mougey, chief chemist and assistant technical director, General Motors Research Laboratories.